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Characteristic Time Correlation of Emissions from Conventional Aircraft Type Flames

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CHARACTERISTIC TIME CORRELATION OF EMISSIONS FROM CONVENTIONAL AIRCRAFT TYPE FLAMES

Environmental Protection Agency
Grant No. R-802650-02 - Final Report

by

J. H. Tuttle, M. B. Colket and A. M. Mellor

Report No. PURDU-CL-76-05

THE COMBUSTION LABORATORY
SCHOOL OF MECHANICAL ENGINEERING
PURDUE UNIVERSITY
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ABSTRACT

The overall combustion process occurring within a liquid spray fueled burner is analyzed in terms of the ongoing dominant subprocesses, with particular emphasis on those subprocesses deemed most critical to pollutant emissions. Liquid fuel evaporation, turbulent mixing, and chemical reaction are each considered separately and are characterized by time scales which typify the importance of each subprocess.

An axisymmetric burner consisting of a flame stabilized in the wake of a disc with a liquid fuel spray injected into the wake region from the center of the disc is considered experimentally. The basic flame structure behind the disc is composed of a hollow reaction region (shear layer) along the boundary between the recirculation zone and the free stream. However as the heterogeneous properties of the combustion process increase in importance, a free stream flame zone following the fuel spray is also apparent.

Guided by the model of the flame structure, the developed characteristic times are combined to form burner output correlating parameters. The success of these parameters is demonstrated by the correlation of both carbon monoxide and oxides of nitrogen exhaust emissions from the disc burner for various geometries, a wide range of burner operating conditions, and two non-similar fuels. In addition since carbon monoxide and oxides of nitrogen emissions originate primarily in separate regions of the flame structure, analysis of the characteristic times reveals a means of optimizing the burner geometry to minimize total pollutant emissions.

The developed characteristic time model is extended to a conventional gas turbine combustor, GT-309. The model predicts the effect of changes in both combustor inlet conditions and combustor geometry on exhaust emissions and is used to demonstrate the design of a low NO_x burner of the GT-309 class.

I. INTRODUCTION

A gas turbine combustor, like any other burner, is a device which converts the energy contained in the chemical bonds of a fuel into sensible enthalpy. However, specific applications of gas turbine engines (particularly aircraft usage) and the engine's basic design scheme have placed rather stringent requirements upon the engine's combustion system. These special requirements have guided the evolution of the present day conventional gas turbine combustor and differentiated it from other combustion devices.

Since the power output of a gas turbine engine is directly related to the thrust produced, the use of a high mass flow rate of air through the engine and also through the combustor is necessitated. Fuel must be mixed with the air and the resulting mixture burned to a high combustion efficiency within a limited volume. The prepared fuel-air mixture must also be such that stable combustion and easy ignition exist at all engine operating conditions. In addition, the combustor must operate at all times with overall equivalence ratios (fuel-air ratio normalized by the stoichiometric fuel-air ratio) much less than unity; otherwise, the temperature of the burned gases entering the turbine component of the engine will exceed the temperature limitations of the materials.

The conventional gas turbine combustor shown schematically in Fig. 1-1 is designed to satisfy all of the design requirements listed above. High mass flow rates of air through the combustor are possible due to the splitting or staging of the compressor discharge air flow into the combustion region. This is accomplished by means of a liner or flame tube whose effective surface hole area distribution determines the mass of air flowing into the combustion region at a particular axial location. It is this axial air flow split which allows the gas turbine combustor to be conceptually divided into three zones, primary combustion, secondary combustion and dilution.

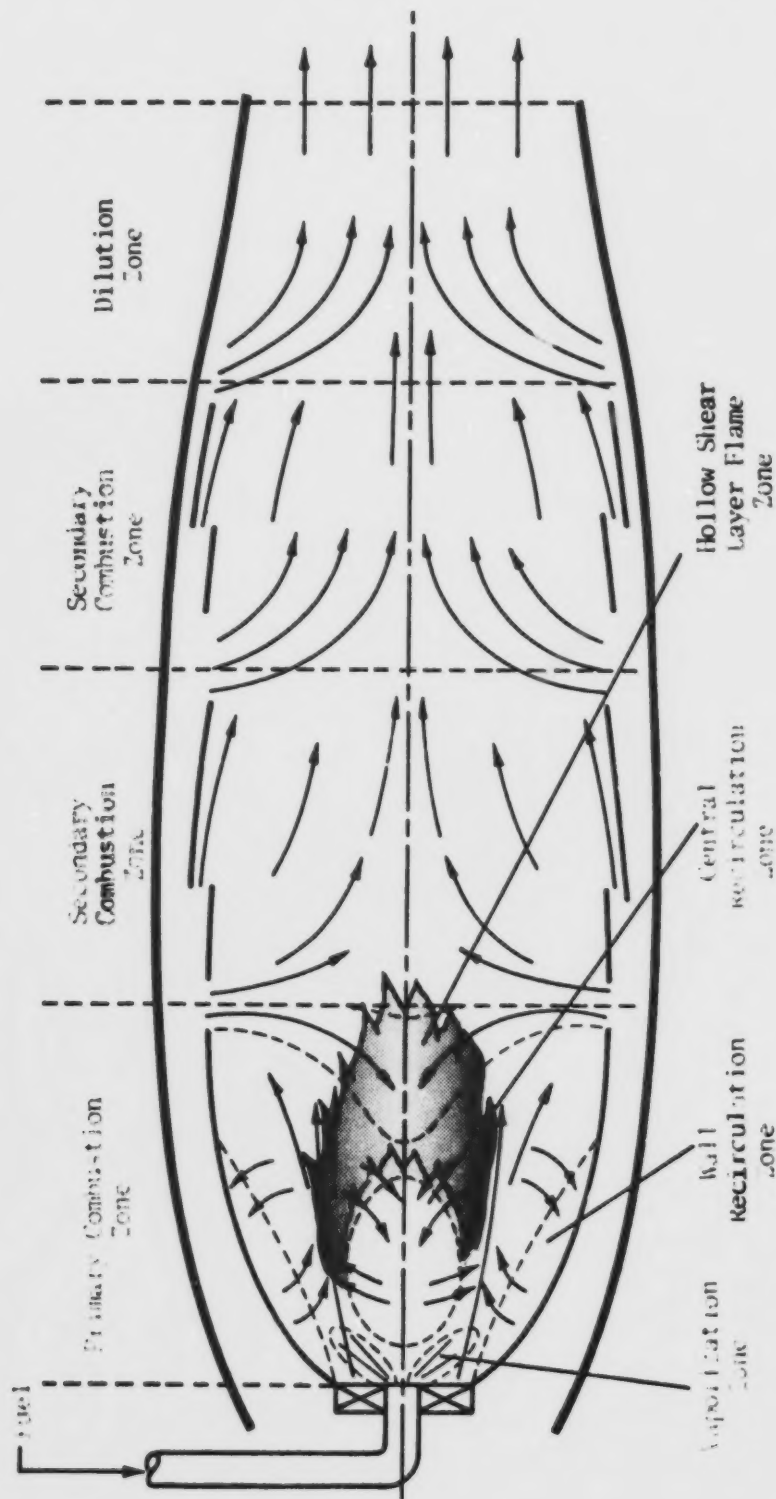


Figure 1-1. Typical gas turbine combustor flow schematic

The primary combustion zone of a gas turbine combustor (Fig. 1-1) normally extends from the fuel nozzle face to the first row of air addition holes and is the region of most intense combustion. The air flow split is such that typical primary zone overall equivalence ratios are near unity which gives the combustion device excellent ignition and stability characteristics. Contained within the primary zone is a region of liquid fuel vaporization as well as regions of recirculating flow. Normally the air flow through the dome of the combustor is swirled which creates a low pressure central recirculation zone for flame holding. Along the boundary between the higher velocity dome air flow and the lower velocity central recirculation zone flow, a region of high turbulence (the shear layer) is created due to the large velocity gradient. The cone angle of the hollow fuel spray is generally such that the liquid droplets penetrate directly into the shear layer region. It is here in the highly turbulent flow of the shear layer that fuel vaporization, mixing of the fuel and oxidizer and burning of the resulting mixture occurs. Both the internal combustor studies of Mellor et al. (1972) and Tuttle et al. (1973) have experimentally observed this hollow shear layer flame structure which surrounds the central recirculation zone and extends through the primary zone of Fig. 1-1.

The air dilution jets through the liner holes marking the beginning of the secondary combustion zone of Fig. 1-1 may penetrate to the combustor centerline and in part bend upstream to feed the low pressure central recirculation zone. The secondary zone of the combustor serves primarily to complete the combustion of fuel vapor or large fuel droplets which were not consumed within the primary combustion zone. These combustible species are pushed to the combustor centerline and are mixed with the secondary air flow. Both Mellor et al. (1972) and Tuttle et al. (1973) experimentally observed the hollow flame structure to close to the combustor centerline in the secondary combustion zone.

The final downstream combustor volume is termed the dilution zone. Its purpose is to reduce the combustor exit plane bulk temperature through air addition to an acceptable turbine inlet temperature. The air addition occurs through both air addition holes and film cooling slots.

The air addition holes produce strong air penetration jets which generate strong mixing and lessen the exhaust gas temperature gradients. Air which enters the combustor through film cooling slots flows along the inner combustor wall and serves as a buffer layer between the hot combustion products and the liner wall.

The design of the conventional combustor described above yields combustion efficiencies in excess of 99% routinely at engine design power operation but somewhat lower near engine idle. In addition, the fixed air addition geometry provides sufficient flow reversal and flame stability over the entire engine load range. Unfortunately, however, the complexity of the ongoing physical processes within a conventional combustor as yet prohibits a detailed numerical simulation of the overall combustion process (but see Mosier and Roberts, 1974).

The internal combustor results of Mellor et al. (1972) and Tuttle et al. (1973) for an Allison J-33 combustor (shown in Fig. 1-2) demonstrate the necessity of correctly modeling the combustor flow properties in order to understand the nature of gas turbine exhaust emissions. The most important features of the J-33 flow model are the fuel rich zone along the dome and liner walls where CO and NO are formed and the strong backflow of air which quenches CO oxidation in the centerline recirculation zone. In particular the authors noted that the lack of substantial air addition into the fuel rich region and too much air addition into the recirculation zone are the reasons for the relatively high emissions of CO from the J-33 combustor. The most significant NO formation was found to occur in the secondary combustion zone where the hollow shear layer flame closes to the combustor centerline. Thus any numerical simulation of the J-33 combustor must include all of the three dimensional flow effects outlined in Fig. 1-2 in order to predict pollutant emissions levels of this combustor since each of the regions is critical to either pollutant formation or destruction.

Present day analytical combustor models are unable to adequately predict the flow pattern within a conventional gas turbine combustor and hence have had limited success in predicting burner output properties (Mellor, 1976). Thus the flame tube designer has necessarily relied

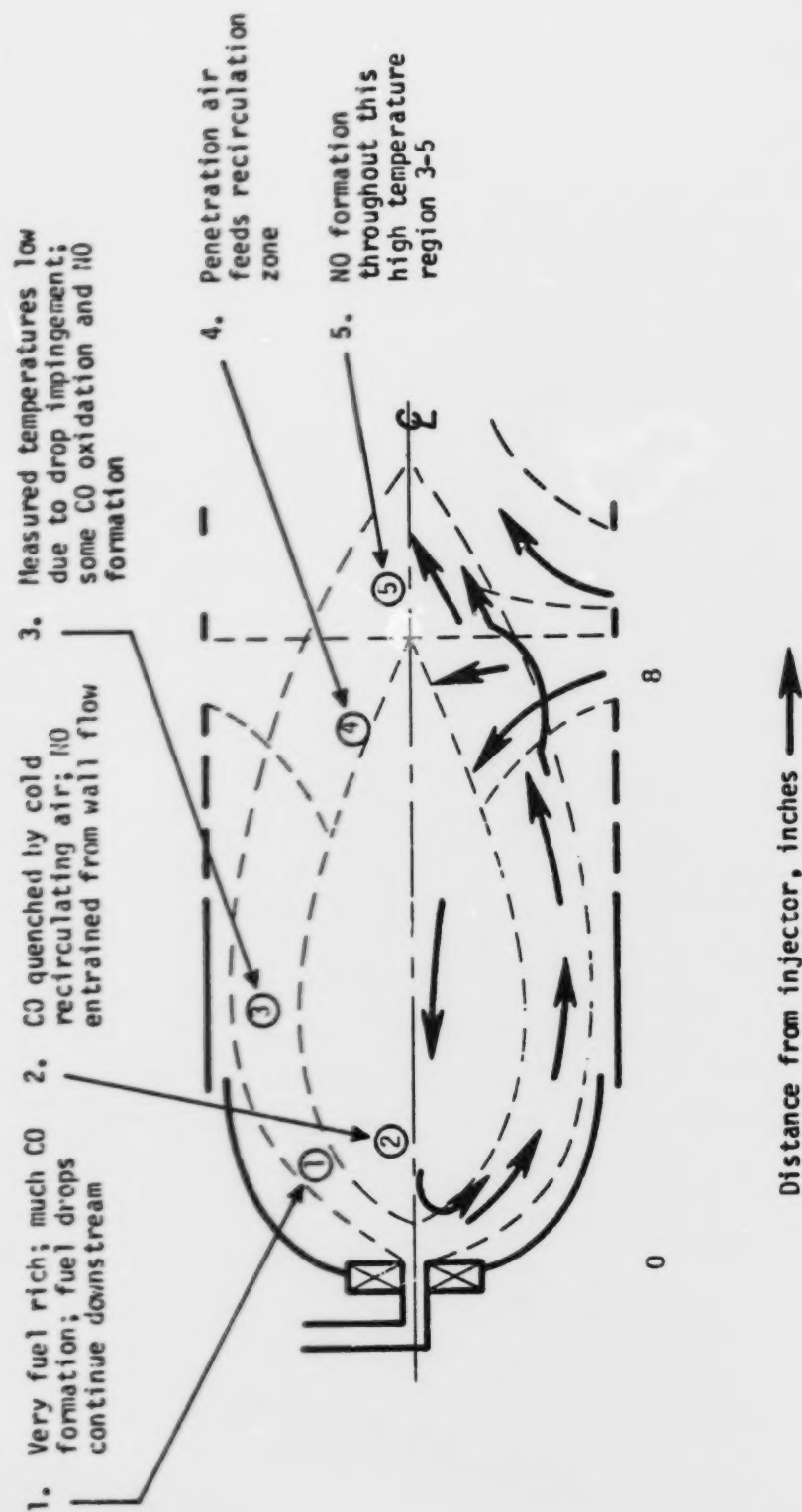


Figure 1-2. Schematic of postulated primary and secondary zone flow pattern for the J-33 combustor (from Mellor et al., 1972 and Tuttle et al., 1973)

heavily upon extensive experimental development programs. An example of such a combustor development program is the work of Cornelius and Wade (1970) who investigated the influence of combustor design on nitric oxide emissions. These authors tested various modifications to a standard GT-309 gas turbine combustor in an effort to reduce the emissions of this pollutant. Although modifications were found which substantially reduced the NO emissions, Cornelius and Wade (1970) did not develop any methods of correlating their measurements against a design modification parameter, but only listed results in terms of percent of the design engine speed. If such a correlating parameter were developed, the parameter would guide the combustor designer in incorporating future liner modifications for further NO reduction.

For a fixed combustor liner design, typically burner output measurements are analyzed in terms of single parameters such as loading, inlet air temperature, atomization properties of the liquid fuel spray, etc., which may be interrelated. An example of a single parameter (combustor pressure) in analyzing a burner output measurement (combustion efficiency) is shown in Fig. 1-3 which is taken from Lefebvre (1968). Identified in Fig. 1-3 are four regimes of pressure. In the first regime, corresponding to pressures of a few psia, Lefebvre (1968) states that good combustion is impossible due to the effects of poor fuel atomization and heat losses. In the second regime, which extends up to 20 or 30 psia, the level of combustion efficiency is largely determined by chemical reaction rates, with atomization playing only a secondary role. The third regime corresponds to normal combustor operation and covers a range of pressures in which the combustion efficiency is close to 100%. Lefebvre (1968) also notes that with increasing pressure, the level of combustion efficiency becomes less dependent upon reaction rates and more dependent on the ability of the combustion zone to evaporate the fuel and achieve the necessary mixing between the fuel vapor, air and combustion products. Thus the author shows a fourth regime in Fig. 1-3 where combustion efficiency may fall off due to limitations imposed by evaporation or mixing. Although combustion efficiency is associated with combustor pressure, in actuality the ongoing fundamental processes of fuel

evaporation, mixing and chemical reaction are determining the combustion efficiency of the burner.

Each of the above discussed methods of analyzing burner output measurements in terms of single engine parameters has been used successfully in the past for limited ranges of burner operating parameters and combustor modifications. However, more useful correlating parameters would be ones which are capable of including over an extended range of burner operation all of the single parameter effects. Once one has quantitatively expressed the controlling physical processes by a combined correlating parameter, it should be possible to make controlled design modifications to improve combustion efficiency and minimize pollutant emissions. The development of these more universal correlating parameters has been the goal of the present research.

In later sections, the sought after burner output correlating parameters are developed from a characteristic time model of the overall combustion process. In this approach, the dominant physical processes in a liquid fueled burner (fuel evaporation, mixing and chemical reaction) are characterized by time scales which typify the importance of each process. Any interaction of the ongoing physical phenomena can be analyzed via combinations of the time parameters, such as ratios or sums. The variation with combustor operating conditions of the characteristic times within the correlating parameters allows the dominant physical processes to control the magnitude of the correlating parameters over the range of engine operation.

The utility of the characteristic time approach in analyzing burner output properties has been demonstrated by others and several of the important investigations in regard to gas turbine performance are briefly discussed below. Danköehler (1940) was one of the first exponents of the use of characteristic times in the analysis of combustion processes. Although much of Danköehler's work is applicable only to homogeneous, pre-mixed processes, a turbulent form of Danköehler's second similarity group is pertinent to gas turbine combustors. This group is the ratio of a turbulent diffusion time to a reaction time. In a gas turbine combustor, this ratio could be used to analyze the time required for preparing a

combustible fuel-air mixture with the time required to burn the resulting mixture. If the ratio were found to be either substantially larger or smaller than unity, then one process is significantly faster than the other. The slower process is then the controlling one of the two (i.e., rate limiting) and emphasis must be placed upon it in the analysis of burner output properties.

The carbon formation process within a conventional gas turbine combustor was analyzed by Heywood et al. (1971). Noting that soot formation occurs in hydrocarbon rich regions within the combustor, the authors analyzed the stoichiometry of the volume of the combustor adjacent to the fuel injector. Three time scales were considered in the analysis: 1) droplet evaporation time, 2) droplet deceleration time, and 3) turbulent fluctuation time. A comparison of the calculated times indicated that significant fuel drop evaporation will take place during the initial deceleration process but a large number of droplets might reach nearly stagnant conditions without vaporizing. The authors note that these droplets coming nearly to rest in the turbulent flow will not burn until the droplet is almost completely vaporized due to the turbulent velocity fluctuations which the larger drops cannot follow. Based on their time scale analysis, Heywood et al. (1971) concluded that a very fuel rich region exists near the fuel injector face. Most of the carbon formation occurs in this region and along the fuel spray trajectory where the equivalence ratios are the highest.

The first attempt to correlate burner pollutant emissions via time scales is the work of Mellor (1973). Recognizing the importance of the fuel droplets in determining the flame structure, Mellor (1973) was able to qualitatively explain the nature of NO emissions of the Allison J-33 combustor with a parameter he interpreted as the ratio of a fluid mechanic time to the fuel droplet lifetime. Mellor (1973) notes that the NO emissions of the J-33 combustor are dependent upon a competition between the presence of fuel droplets and the homogeneous kinetic production of NO. At combustor operating conditions known to have good fuel spray atomization (short fuel droplet lifetime) NO emissions correlated well with the fluid mechanic time only. However as the fuel injector efficiency deteriorated, the effect of fuel droplets upon the NO emissions

must be taken into account. In recalling the internal flow field of this combustor (Fig. 1-2), the effect of increasing the fuel droplet lifetime is to lengthen the shear layer flame. Thus a greater percentage of the total fuel flow enters the secondary combustion zone and increases the intensity of the combustion process in region 5 of Fig. 1-2, and this is the region where the earlier investigations of both Mellor et al. (1972) and Tuttle et al. (1973) showed NO formation to be most significant. Hence, the magnitude of the fuel droplet lifetime primarily influences the combustor NO production through the distribution of fuel within the overall flame structure.

Swithenbank et al. (1973) analyzed several combustor performance variables (including stability and combustion efficiency) in terms of time parameters representing the fuel drop lifetime, the mixing time, the chemical reaction time and a combustor stay time. A geometric combination (square root of the sum of the squares) of the fuel droplet lifetime and the turbulent mixing time was taken as an effective mixture preparation time. This effective mixture preparation time was then compared to the stay time of the gases within the flame region of the combustor to determine the unmixedness of the burning gases. Swithenbank et al. (1973) found that the principle effect of increasing the unmixedness was to lower the conversion of fuel to products and thus decrease combustion efficiency.

Calculations of characteristic times for fuel droplet evaporation, mixing of fuel and air, combustion and thermal NO formation were published by Vranos (1974). Based on the magnitudes of these calculated times, the author was able to qualitatively explain the nature of NO_x emissions from liquid fueled gas turbine combustors operating near design. The burning time of the fuel-air mixture was found to be significantly lower than the other three time scales. Thus the author concluded that the rate of fuel burnup is controlled by the fuel evaporation and mixing times, which are of similar magnitude. As a result, Vranos (1974) concluded that NO formation is primarily dependent on the amount of fuel burned since the local mixtures and local temperatures are fixed by the mixing rate. In addition, these local mixtures must be near unity equivalence ratio, because only there are the local temperatures of

sufficient magnitude to form appreciable NO. The conclusions of Vranos (1974) are in agreement with the present work and will be expanded upon in the next section.

A more recent use of time scales is the work of Mellor (1976) who discussed methods of reducing pollutant emissions from gas turbine engines in terms of variations of characteristic times. Time scales similar to those of Vranos (1974) were used except Mellor (1976) noted that there are two mixing processes of importance within the primary zone of a conventional combustor: one involving the mixing of fuel vapor eddies with hot recirculated burned gases, and the other characterizing the mixing of the fuel-burned gas mixture with fresh air in the shear layer flame region. Mellor (1976) discussed techniques of varying each characteristic time and what the resulting effect of the time scale variation would be upon pollutant emissions. For example, Mellor (1976) noted that improving the liquid fuel spray atomization will decrease the fuel droplet lifetime which in turn would reduce HC and CO emissions by providing fuel vapor at a faster rate and would reduce NO emissions by decreasing the overall flame length. In addition, the author analyzed present Environmental Protection Agency (1972) suggested methods of combustor modification to reduce emissions in light of the introduced characteristic times.

Several of the above mentioned time scales are further quantified in the present investigation. In Section II the dominant physical processes occurring within a liquid spray fueled burner (fuel evaporation, turbulent mixing, and chemical reaction) are considered in detail and are depicted by appropriate time parameters which typify the importance of each process. Not only will the effect of each time on the overall combustion process be included, but also methods of estimating the magnitude of these time parameters.

The experimental combustion facility and its operation are topics of Section III. The experimental apparatus was used to obtain pollutant emissions measurements from a simplified, axisymmetric diffusion flame burner whose operation and geometry could easily be varied. The utility of the characteristic times is demonstrated in Section IV where the burner

exhaust measurements are correlated against parameters developed from the time scales for a wide range of burner operation, various geometries, and two non-similar fuels. In addition, an analysis of the characteristic time parameters is given which shows an optimum burner geometry exists for minimum emissions.

In Section V, the emissions correlating parameters developed for the simplified, axisymmetric burner are extended to a conventional automotive gas turbine combustor. Semi-empirical expressions for carbon monoxide and oxides of nitrogen emissions indices (COEI and NO_xEI , respectively) as functions of mixing and kinetic times are developed from combustor exhaust emissions data (Cornelius and Wade, 1970; Wade and Cornelius, 1972; Cornelius, 1976) for variations in compressor/regenerator discharge conditions and combustor geometry. These equations are then used to predict near design and idle emissions for differing combustor geometries (the combustor inlet conditions are usually fixed by the engine cycle design and are thus not variable to the burner designer), and a low NO_x burner is suggested as a sample of the resulting optimization process. Finally, conclusions drawn from the present use of characteristic times in the development of correlating parameters for burner output measurements are stated in Section VI. In the present work, we have attempted to show how such correlations can eliminate some of the cut-and-try development currently required for combustor design. Extension of the present method of correlating engine emissions to aircraft combustors will allow the combustor designer to make controlled burner modifications to improve combustion efficiency and minimize pollutant emissions.

II. CHARACTERISTIC TIMES

The combustion of liquid fuel sprays in a realistic burner is an extremely complex phenomenon, involving the simultaneous processes of heat, mass, and momentum transfer in a reacting turbulent flow. Parameters controlling the combustion process are expected to include fuel characteristics, fuel drop size, spatial distribution of the fuel droplets, and the aerodynamics of the combustion device. In the present approach, the overall spray combustion process is analyzed in terms of the various, but important subprocesses which determine the flame structure and hence the burner emissions characteristics.

In liquid fuel spray flames, three dominant subprocesses must occur before combustion of the fuel is complete. These three are 1) vaporization of the liquid fuel droplets, 2) mixing of the vaporous fuel with oxidizer to yield a combustible mixture, and 3) combustion of the resulting fuel-air mixture. Following Mellor (1976) (Table 2-1), time parameters will be used to characterize these three subprocesses which control conventional gas turbine combustion and pollutant formation phenomena. Extensive use will be made of the subscripted symbols listed in Table 2-1 throughout the text.

The first time listed in Table 2-1, the fuel droplet lifetime, is of paramount importance in determining the heterogeneous properties of the combustion process and is of course associated with the fuel spray and vaporization region of Fig. 1-1.

The second and third time parameters of Mellor (1976) are both associated with the turbulent mixing properties of the reacting flow field. τ_{fi} , which characterizes the mixing process near the fuel injector of Fig. 1-1 is expected to be of increasing importance in combustors utilizing air blast or air assist nozzles, rather than pressure atomizers. With either air blast or air assist nozzles, unburnt, high velocity air is injected along with the fuel. The high velocity air not only shatters

Table 2-1

Characteristic Times for Combustion and
Pollutant Formation in Two-Phase Turbulent Flow
(from Mellor, 1976)

<u>Time</u>	<u>Symbol</u>	<u>Physical or Chemical Processes</u>
Fuel droplet lifetime	τ_{eb}	Droplet evaporation and/or combustion
Eddy dissipation time for injected fluid	τ_{fi}	Small scale turbulent mixing near the fuel injector in the recirculation zone
Eddy dissipation time in the shear layer	τ_{sl}	Large scale turbulent mixing between fresh air and the recirculating burned gas - fuel mixture
Fuel ignition delay and burning time	τ_{hc}	Homogeneous combustion of the fuel to CO_2
NO formation time	τ_{no}	Homogeneous kinetics for NO formation

the fuel flow into small droplets but also provides a source of oxidizer within the recirculation zone. τ_{fi} then characterizes the subsequent mixing of fuel vapor and the provided oxidizer. If τ_{fi} is short (rapid mixing), then the extremely fuel rich regions near the fuel spray which are predominantly responsible for soot generation, tend to be leaned out. Norster and Lefebvre (1972) as well as numerous other authors have noted that a large decrease in smoke emissions of gas turbine combustors can be realized through use of air blast rather than pressure atomizers.

For combustors utilizing a fuel pressure atomizer, τ_{fi} is of limited importance because the recirculation zone contains only oxygen deficient gases. Since only pressure atomizers were considered in the present experimental investigation, τ_{fi} will not be discussed further.

The turbulent mixing occurring in the shear layer region (see Fig. 1-1) is characterized by τ_{sl} and is associated with all combustors which have a recirculating flow region. This time determines the rate of mixing between the fresh incoming air and the recirculating burned

gas-fuel mixture and is dependent upon the scale and intensity of the turbulence in the shear layer region.

The last two times listed in Table 2-1, τ_{hc} and τ_{no} , are scales which represent the times required for homogeneous combustion of the fuel and NO formation, respectively. These processes take place after mixing of the fresh air and fuel on the molecular level and are primarily dependent upon the local mixture ratio, combustor pressure, and local temperature.

With the exception of τ_{fi} , each of the times listed in Table 2-1 is discussed in detail below. Not only will the effect of each time on the overall combustion process be included, but also methods of estimating these time parameters.

A. Heterogeneous Processes

Mellor (1973) showed that for a gas turbine combustor operating near idle, poor atomization of the liquid fuel spray could result in increased emission indices of CO, HC, and NO. Hence in order to correlate burner emissions over a wide range of operating parameters, the atomization properties of the fuel nozzle must be considered over the same range. The atomization efficiency at a given operating point will determine the importance of heterogeneous processes in the overall combustion process.

Fuel droplet evaporation begins when the fuel droplet comes into contact with hot gases. In a combustor, this process is accelerated by recirculating hot burnt gases into the vicinity of the fuel injector. Since the burnt gases are oxygen deficient products of combustion, significant combustion is not possible within the vaporization zone of Fig. 1-1. Rather the hot gases vaporize the fuel and convect the resulting fuel vapor outward along the outer boundary of the recirculation zone where mixing between the fuel vapor and oxygen-rich dome air flow begins to form the shear layer flame zone.

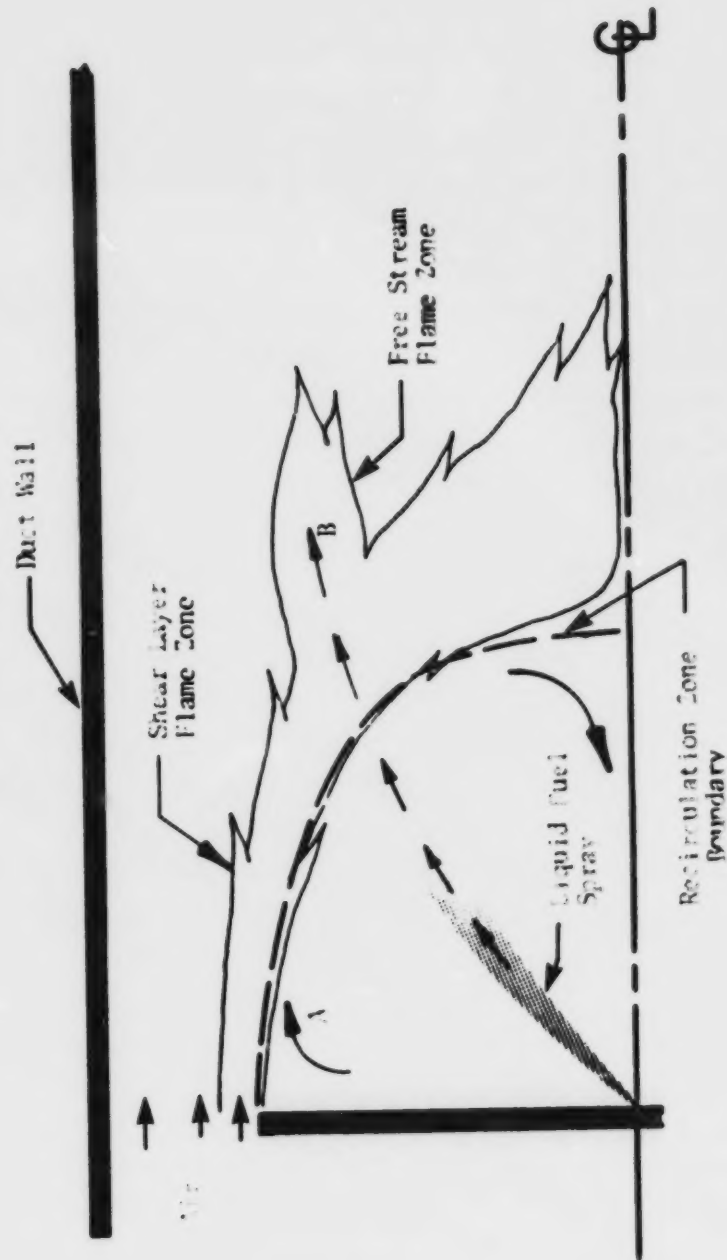
The magnitude of the fuel droplet lifetime, τ_{eb} , will affect the basic flame structure and pollutant emissions of a conventional combustor. If τ_{eb} is short, a high percentage of the liquid fuel is

vaporized early in the combustion process which reduces the flame length. In this case provided dome air flow is sufficient, most of the fuel is consumed early in the flame and within the primary combustion zone, and hence the hollow shear layer flame would not close to the combustor centerline in the secondary combustion zone with the intensity noted by Tuttle et al. (1973) for the J-33 combustor. However, if the droplet lifetime is excessively long, the majority of the fuel droplets will penetrate far into the flow field. This slower rate of vaporous fuel addition lengthens the flame and greatly increases the importance of the secondary combustion zone in the total combustion process. Thus the magnitude of τ_{eb} determines the importance of the secondary combustion zone to overall engine emissions.

Since the recirculation zone is positioned downstream of the fuel spray in a gas turbine liner, most of the fuel as mentioned earlier enters the shear layer flame zone. Thus with the combustion of all fuel particles (both large and small droplets) taking place throughout the same region of the combustor, it is somewhat difficult to determine whether or not the large droplets within a given fuel spray play a role in determining combustor performance.

In the present investigation, the experimental burner was designed to separate the large and small droplet effects and is shown schematically in Fig. 2-1. This axisymmetric burner consists of a disc centered in a duct, with a recirculating flow region for flame stabilization behind the disc due to blockage of the duct air flow by the bluff body. A hollow cone liquid fuel spray is injected directly into the recirculation zone. Vaporized fuel and small droplets follow the recirculating flow and are carried outward along the disc to feed the shear layer flame (Fig. 2-1). The larger drops tend to maintain the spray sheath and travel directly to the shear layer. Thus the disc stabilized flame is ideally suited for studies pertaining to the influence of fuel spray atomization upon liquid spray combustion.

The magnitude of the fuel droplet lifetime, τ_{eb} , as in the case of a turbine combustor, will determine the basic flame structure behind the disc stabilizer. If the droplet lifetime is short, a high percentage of the liquid fuel is vaporized within the recirculation zone and



A - Path of small droplets and fuel vapor
 B - Path of large droplets

Figure 2-1. Flow separation of large and small fuel droplets

subsequently convected towards the disc edge to feed the shear layer flame zone. However if the droplet lifetime is long, a significant amount of fuel may penetrate through the shear layer into the free stream. In this latter case, the flame structure may exhibit a free stream flame zone (Fig. 2-1) which follows the trajectory of the fuel spray. Since the magnitude of the droplet lifetime determines the relative importance of the two flame zones depicted in Fig. 2-1 and these two flame zones do have different pollutant emissions characteristics, any parameter capable of correlating combustor emissions must include a functional dependence upon τ_{eb} .

The accepted physical relationship which governs either droplet evaporation or combustion is the " d^2 law" of Godsave (1953).

$$d_f^2 = d_o^2 - \beta t \quad 2-1$$

where d_f = instantaneous fuel drop diameter
 d_o = initial drop diameter
 β = evaporation or combustion coefficient
 t = time.

The characteristic time for droplet evaporation or combustion is obtained by letting d_f go to zero. From Equation 2-1, the droplet lifetime, τ_{eb} , is seen to be

$$\tau_{eb} = d_o^2 / \beta \quad 2-2$$

β (defined by Equations 2-3, 2-4, and 2-5 below) is a function of fuel and ambient gas properties. For convective atmospheres, β is also a function of the Reynolds number (based on the mean drop size and drop relative velocity) and the Schmidt number. Mellor (1972) has summarized several empirically determined relationships for the dependence of β upon the Reynolds number and the Schmidt number.

$$\beta = \beta_o (1.0 + f(\text{Re}, \text{Sc})) \quad 2-3$$

$$\beta_o = \frac{8k}{\rho_l c_p} \ln (1 + B) \quad 2-4$$

$$B = \frac{c_p (T - T_l)}{L} \quad 2-5$$

where β_o = evaporation coefficient in a stagnant atmosphere

k = gaseous thermal conductivity

ρ_l = density of liquid fuel at T_l

c_p = gaseous specific heat at constant pressure

B = transfer number for droplet evaporation

T = temperature of ambient gas

T_l = boiling point of liquid fuel at the local pressure

L = sensible enthalpy of the liquid fuel from the initial temperature to T_l plus the latent heat of evaporation.

In the present investigation, β was corrected for forced convection by using a correlation due to Frössling (see Ranz and Marshall, 1952).

$$\beta = \beta_o (1.0 + 0.276 \text{Re}^{1/2} \text{Sc}^{1/3}) \quad 2-6$$

The initial droplet diameter must be determined either by measurement or from empirical correlation before τ_{eb} can be calculated. However, since no practical fuel atomizer will produce a spray of uniform droplet size, reliance upon a mean size is necessary. For purposes of combustion or evaporation, a special mean, the Sauter mean diameter (SMD) is most often used. The SMD is defined as the diameter of a droplet whose volume to surface ratio is the same as the volume to surface ratio of the entire spray. Thus the SMD of the spray is inversely proportional to the spray surface to volume or mass ratio. The SMD is then actually an indicator of the total surface area per unit mass of fuel, or a measure of the degree of atomization. Here the Sauter mean diameter of the spray was used as the initial drop diameter whereas the fuel droplets

penetrating farthest downstream are those that have an initial diameter much larger than the SMD.

Several fuel injector investigations in which empirical relations for determining the spray SMD were included are summarized by Tuttle et al. (1975). In the present work, the initial drop diameter was calculated following the results of Hunter et al. (1974) for simplex pressure nozzles.

$$SMD = \frac{90.5 \nu^{.3} \rho^{.205} Q^{.205}}{\Delta P^{.354}} \quad 2-7$$

where SMD = Sauter mean diameter, micron

ν = viscosity, centistoke

ρ = density, gm/cm³

Q = volumetric flow rate, lit/hr

ΔP = nozzle differential pressure, atm.

As can be seen from Equation 2-2, the magnitude of τ_{eb} is determined by the initial fuel droplet diameter and the evaporation coefficient. Hence to vary τ_{eb} at a constant burner operating condition (i.e. no change in overall equivalence ratio, air flow rate, inlet air temperature, or combustor pressure) will require either a change in fuel injector or a change in fuel type. Both of these methods were used in the present experimental investigation which is the topic of Section III.

B. Mixing Processes

In the primary combustion zone of a gas turbine liner, rapid mixing of fuel and air is a necessity in order to achieve the high heat release rate per unit volume required by aircraft engines. However the energy available for turbulent mixing is directly proportional to the pressure drop across the liner which typically does not exceed 4% because of the detrimental effect upon engine cycle efficiency. Thus the mixing rate is limited and stirred reactor conditions are not realized within conventional gas turbine combustors.

Insight into the flow pattern within combustors was provided by the internal combustor measurements of Mellor et al. (1972), Tuttle et al. (1973), and Shisler et al. (1974). These authors each found the gas turbine primary combustion zone to have a combustion mechanism resembling a turbulent diffusion flame in the region of the shear layer (Fig. 1-1). In this type of flame structure, one can visualize two adjacent streams of turbulent fluid, one pure fuel and the second pure air. Along the interface of the two streams, an interlapping of fuel-rich and air-rich eddies exists, the extent of which has been labeled "unmixedness" by Hawthorne et al. (1949) and "intensity of segregation" by Brodkey (1967). This phenomenon of unmixedness limits chemical reaction rates below those indicated by mean concentrations and is discussed further in the work of Bilger (1976). These pockets of fuel and air continue to dissipate and the chemical reaction occurs at or closely about a stoichiometric fuel-air mixture (see for example, Wohl et al., 1949). The reaction rate of the resulting mixture can be considered fast in comparison to the mixture preparation time. Thus the location and width of the flame zone is determined primarily by the mixing processes and not by the chemical rates. Since the flame structure is heavily dependent upon the mixing process, burner emissions are expected to show a similar strong dependence upon a characteristic mixing time.

In addition to the turbulent mixing of fuel and air to establish the flame structure, other mixing processes within a gas turbine combustor can be identified. Hot combustion products from the downstream portion of the flame structure are mixed into the recirculating flow and convected upstream. These hot gases are in turn mixed with and serve to ignite the fresh fuel-air mixture in the upstream portion of the flame structure. Thus a stable combustion system results which is self-ignited without the aid of an external energy source such as a pilot flame or spark plug. Although these mixing processes involving the centerline recirculation zone are critical to combustor stability, their importance is diminished in relation to combustor pollutant emissions.

The last mixing process to be discussed here results from the addition of dilution air along the liner wall for either film-cooling or jet mixing. If the cooling air is added too rapidly, the resulting mixing between cold air and hot combustion gases will effectively cool the hot gases to the extent that chemical reactions are frozen. The most rapid quenching of chemical reactions is expected to occur around strong air dilution jets which penetrate far into the combustor (Morr et al., 1975). In general, these jets will enter the liner normal to the main combustor gas flow and will thus produce high fluid shearing regions and rapid mixing. Air entering the combustor through film-cooling slots will have a lesser effect upon quenching since this air flows parallel to the main combustor gas flow.

After the initial oxidation of the hydrocarbon fuel to form CO, this rather stable combustion intermediary may exist in quite large (near equilibrium) concentrations in the flame zone. Ideally, this species burns out completely in the fuel lean regions on the air side of the flame. However if cold air jets penetrate into the regions of CO burnout, CO oxidation reactions are effectively frozen in volumes of the combustor where CO concentrations are high. In this case, the engine exhaust CO emissions will be high, reflecting the high concentrations in the regions of quenching. A mixing time relevant to CO emissions is thus one which is appropriate to the rate of mixing of cold liner air with hot gases in the lean regions of the flame.

The internal combustor probing results of Mellor et al. (1972) and Tuttle et al. (1973) discussed earlier showed NO formation to be most significant near the centerline of the combustor downstream of the central recirculation zone where gas temperatures are high and velocities low. The lack of mixing of the hot combustion products in this region of the combustor with neighboring colder species accounts for high combustor NO emissions. A mixing time relevant to NO emissions is then one which characterizes the mixing of colder species with the hot gases in this high temperature region. Note that NO emissions are excessive due to a slow rate of reaction quenching near the combustor centerline whereas too rapid quenching in the air lean portion of the

flame structure produces high CO emissions. Hence a unique mixing time scale should not be expected to be associated with both CO and NO emissions.

The actual turbulent mixing processes within combustors are only partially understood at best. In the development of most qualitative models of mixing, strong analogy between the transport processes of mass and momentum is assumed. The mixing process is assumed to occur by the engulfment of the entering species directly into large scales. These scales in turn cascade down to small scales where molecular diffusion occurs. As in the case of energy dissipation, the relatively long persistence of the large scale eddies is taken as the slow step in the overall mixing process (Gouldin, 1973) and hence is emphasized here.

Present attempts to predict turbulent mixing times have closely followed earlier work in turbulent flows. For example, Corrsin (1957) treated the problem in terms of concentration fluctuations analogous to velocity fluctuations. Using this approach, Corrsin (1957) developed a mixing time which was in actuality the time constant for the exponential decay of the concentration fluctuations. This time constant is the reciprocal of the " β " parameter commonly used in stirred reactor theory to characterize the mixing (see Evangelista et al., 1969) and depends on the mass of fluid to be mixed, integral turbulent length scale, and the power input to the mixer. The applicability of this mixing time (Corrsin, 1957) to the turbulent flow within a combustor is unclear, since the derivation required the assumption of stationary, isotropic turbulence in a non-reacting, constant temperature flow field.

One of the largely unsolved problems connected with reacting flows is the effect of chemical reaction upon turbulent flow (Libby, 1975). Thus the extension of any cold flow treatments of mixing phenomena to reacting flow is questionable. Consequently, here a very simplistic mixing time will be used. As stated above, the slow step in the mixing process is considered to be the breakdown of the initially formed large eddies. Thus a representative mixing time can be taken as the lifetime or dissipation time of these large scale turbulent structures. Following Tennekes and Lumley (1972), the characteristic breakdown time

for large eddies, τ , is

$$\tau \sim L/u'$$

2-8

where L is the integral length scale (largest eddy size) and is generally taken to scale with a geometrical dimension of the flow device. The turbulent (rms) velocity, u' , is a measure of the amplitude of the velocity fluctuations in the turbulent flow and is normally proportional to the mean flow convective velocity. However since the CO and NO_x combustor emissions originate primarily in different regions of the flame structure, care must be taken in selecting the correct L and u' for the region of interest.

The appropriate geometrical combustor dimensions for use as turbulent length scales cannot be totally determined a priori but must be quantified through analysis of experimental emissions data. Also since the dilution air is added at discrete locations along the liner wall, it is anticipated that a unique integral length scale may not be appropriate for all combustor operating conditions due to possible shifting in flame position or flame length over the range of combustor operation. For the simplest case possible, that of a fixed integral length scale, the turbulent mixing time scales are just inversely proportional to the convective velocity. Hence for this case, pollutant emissions should correlate with a representative convective velocity (inverse of time). This is the basis for the correlation of NO, CO, and HC emissions with the reference velocity (defined as the total burner air flow divided by the product of the combustor inlet density and the maximum cross-sectional area), a parameter used successfully by NASA workers for many years. (See, for example, Grobman (1972) and Jones and Grobman (1973)).

With the disc combustor (Fig. 2-1), there is no discrete addition of dilution air through a liner wall; rather the air is added continuously along the shear layer. CO oxidation reactions will be quenched most efficiently near the rim of the disc since there the mixing rate between hot, partially burnt combustion gases and cold inlet air is greatest (highest convective velocity due to the minimum flow

area at the plane of the disc). Thus CO emissions should be strongly related with a mixing time appropriate to the shear layer but near the rim of the disc whereas nearer the centerline of the burner where mixing rates are decreased, there is sufficient time for CO burn-out. For NO on the other hand, very little will be formed in the shear layer near the rim of the disc, most of it being formed in the downstream portion of the shear layer and in particular the region of hot gases near the burner centerline downstream of the recirculation zone. NO emissions should then be coupled strongly with a mixing time appropriate to this centerline region since NO formation will continue to occur there within hot turbulent eddies only until these fluid elements have been sufficiently mixed with colder species. Since CO and NO emissions originate from different regions of the flame structure, a unique shear layer mixing time scale, τ_{sl} , is not appropriate for correlation of both CO and NO_x emissions.

For the outer periphery of the shear layer, the appropriate turbulence velocity scale, u' , will be a function of the velocity through the annulus, V_a

$$V_a = \frac{\dot{m}}{\rho A} \quad 2-9$$

Here $\dot{m}$ is the mass flow rate of air through the burner, ρ is the combustor inlet density, and A is taken to be the flow area at the disc location (area of duct minus area of disc). Then

$$\tau_{sl,co} = L_{co}/V_a \quad 2-10$$

where the subscripts sl and co refer to a mixing time in the shear layer but in the outer portion of the shear layer from which CO emissions are high.

The length scale, L , is more difficult to choose as in the shear layer mixing region it grows linearly with distance from the disc rim (Tennekes and Lumley, 1972). However the rate of dissipation of the concentration fluctuations will depend upon the initial size of the vortices shed from the disc edge and the subsequent growth rate of these fluid

elements. Although the fluid mechanic phenomena leading to the shedding of vortices from an axisymmetric object are certainly more complex than for the case of a cylinder in a cross flow, similar scaling relations are assumed. Since the initial size of the wake eddies scales with the width of an object normal to the freestream flow, the appropriate integral length scale for the disc combustor is taken to be proportional to the disc diameter, d .

The choice of the disc diameter as the appropriate dimension for scaling the initial size of the eddies shed from the disc was experimentally investigated in the present work through use of various disc diameters and through use of toothed disc flameholders. Although the dimensions of the teeth on the toothed discs were small in comparison to the disc diameter, the fact that the teeth protrude into the flow should alter the mixing phenomena in the shear layer region immediately downstream of the disc.

With a toothed disc flameholder, vortex shedding (whose initial size scales with the average disc diameter similar to the circular disc case) will continue. However, additional vortices whose initial size scales with the tooth width, W , will also be generated due to flow around the teeth. Hence the overall mixing process may be accelerated behind a toothed disc flameholder, depending upon the size of the tooth generated vortices relative to those shed by a circular disc. Here the appropriate turbulent length scale is a weighted average of the reciprocals of these two initial eddy sizes.

$$\frac{1}{L_{co}} = \frac{1}{d} + k \frac{1}{W} \quad 2-11$$

Since the two initial eddy sizes only scale with the appropriate geometrical dimensions and the exact scaling laws are unknown, a proportionality constant k which relates the two scaling laws must be included in Equation 2-11. Evaluation of k must await the analysis of the experimental measurements.

For NO formation, the appropriate length scale is also the disc diameter. Since NO formation is most significant near the centerline of the burner downstream of the recirculation zone, the velocity scale

is taken to be the bulk velocity of the flow through the full duct area. Because the bulk flow will accelerate along the shear layer due to heat addition, the duct velocity in the region of interest is somewhere between that for the unburnt and fully burnt flow. If an average velocity is taken, the result is

$$\tau_{sl,no} = \frac{L_{no}}{\frac{V_o}{2} \left(1 + \frac{T_b}{T_i} \right)} \quad 2-12$$

where V_o is the duct velocity of the unburnt gas, T_b is the temperature of the burnt gas, and T_i is the temperature of the inlet air flow to the burner. Note that since NO formation occurs away from the rim of the disc, no change in L_{no} is expected between the circular and toothed discs.

The two mixing times developed above for the disc burner, $\tau_{sl,co}$ and $\tau_{sl,no}$, were varied over wide ranges in the present experiments through changes in both the length scale and convective velocity. The use of these times to guide burner modification for reduced total pollutant emissions (CO and NO_x) will be demonstrated in Section IV.

C. Chemical Processes

Three additional times also inherent to gas turbine combustion have yet to be discussed. These times are associated with the homogeneous chemistry of the fuel/air combustion and the formation or destruction of pollutant species. The first is the fuel ignition delay and burning time, τ_{hc} , and is the time required for the molecular reactions to proceed once the fuel vapor and air are mixed. This time is primarily dependent upon the local equivalence ratio, combustor pressure, and local temperature. Values of τ_{hc} versus overall equivalence ratio were estimated by Vranos (1974) by dividing laminar combustion wave thickness by the laminar burning velocity. For convenience, results of the τ_{hc} calculations of Vranos (1974) are listed in Table 2-2 for a typical gas turbine combustor operating condition. Other means of estimating τ_{hc} from reaction rate data exist in the literature (see for example Dodds et al., 1975) but the calculated results are not too

different from those listed in Table 2-2.

Table 2-2

Values of τ_{hc} (from Vranos, 1974)
($P = 15 \text{ atm}$, $T_i = 700^\circ\text{K}$)

ϕ	τ_{hc} msec
0.7	0.20
0.8	0.17
0.9	0.15
1.0	0.13
1.1	0.16
1.2	0.20
1.4	0.40

As can be seen in Table 2-2, near unity equivalence ratio τ_{hc} is a small fraction of a millisecond. These relatively short times in relation to accepted times for other gas turbine combustion processes are in agreement with the earlier statement that the gas turbine flame structure resembles a turbulent diffusion flame. Thus at stable combustor operation, τ_{hc} is sufficiently smaller than the mixing times that the rate of fuel burnup is limited only by the rate of the mixing processes.

For stable burner operation, the initial reaction of the hydrocarbon fuel to form CO is also fast compared to the subsequent oxidation of CO and the formation of NO. Since these latter two processes are critical to burner emissions levels, here emphasis is placed upon the slower CO oxidation and NO formation times. The ability to describe these two chemical times in terms of the actual kinetic processes is however severely limited by the fluid's unmixedness as described earlier and by unknown kinetic parameters and species concentrations. Thus in this work, an alternative, semi-empirical approach has been employed in identifying these kinetic times.

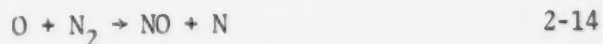
At temperature levels normally present in gas turbine combustors, the reactions responsible for thermal NO formation are slow compared to the fuel oxidation reactions and are particularly temperature sensitive. This strong dependence of NO formation rate upon temperature was utilized by Lipfert (1972) to successfully correlate NO_x emissions from a variety of conventional gas turbine engines. In particular, Lipfert (1972) observed that the oxides of nitrogen are related directly to the combustor inlet air temperatures.

Sawyer et al. (1973) observed that Lipfert's (1972) correlation of NO_x emissions from a number of different gas turbine engines operating over a range of power settings with the combustor inlet air temperature, could be interpreted in terms of a simple kinetic model. Noting that the local temperature is the most important parameter affecting the rate of NO formation, Sawyer et al. (1973) replotted the NO_x emissions data presented by Lipfert (1972) in standard Arrhenius form (Fig. 2-2), $\ln \text{NO}_x$ versus $1/T_f$. The maximum local temperature within the combustors, T_f , was predicted from inlet temperature data under the assumption of stoichiometric combustion. The good correlation exhibited in Fig. 2-2 is another indication of the diffusion flame character of conventional gas turbine combustors, i.e. combustion is occurring at near unity local equivalence ratio.

The correlation of NO_x emissions displayed in Fig. 2-2 can be expressed as*

$$\ln \frac{\text{NO}_x}{\text{constant}} = -135 \text{ kcal/mole}/RT_f \quad 2-13$$

where T_f is the calculated stoichiometric flame temperature. This temperature dependence coincides with that of the rate limiting step for thermal NO formation, i.e.



*Sawyer et al. (1973) suggested an activation energy of 136 kcal/mole. However if the effect of combustor pressure upon T_f is considered, a value of 135 kcal/mole is preferred.

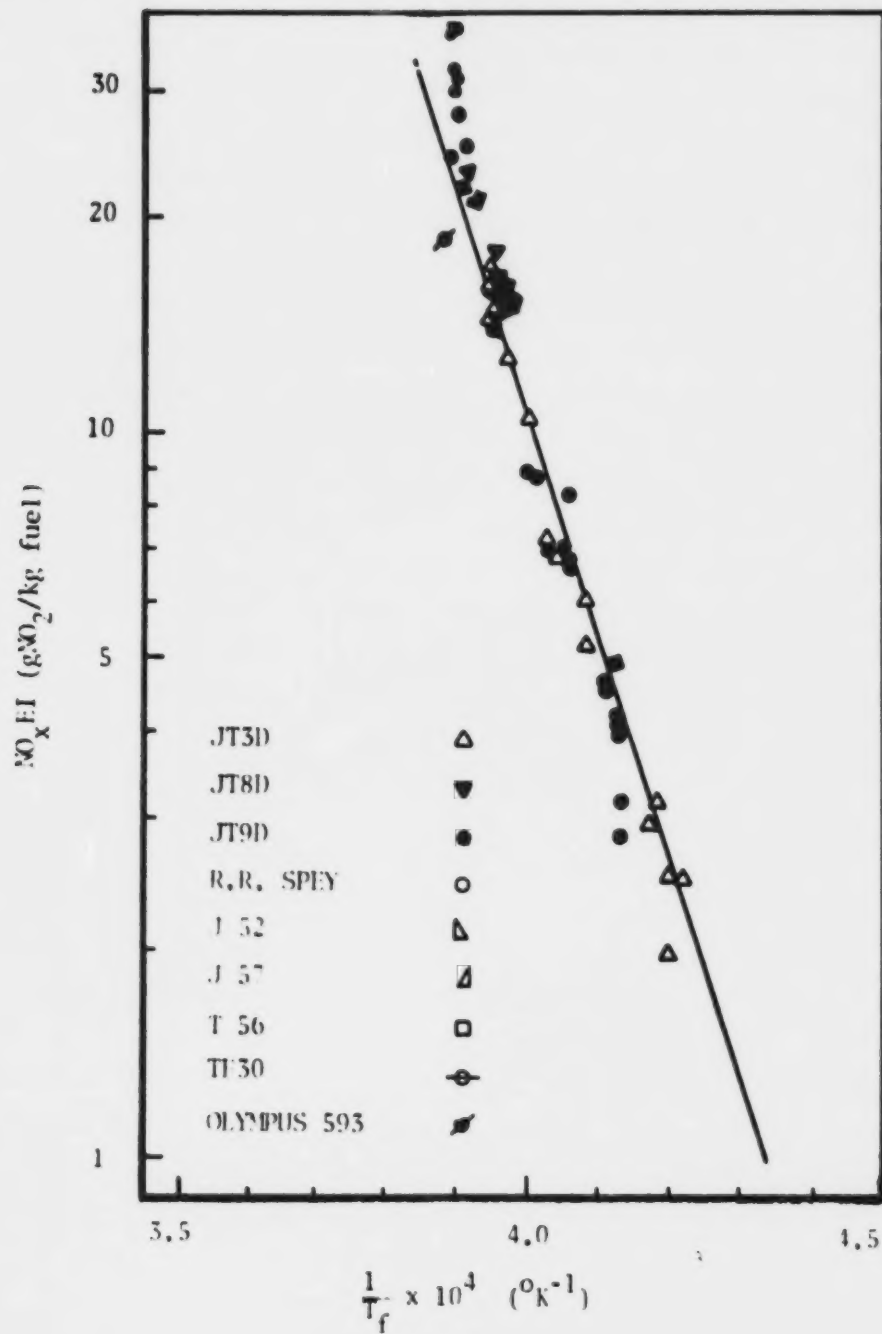


Figure 2-2. Correlation of $\text{NO}_x \text{EI}$ with stoichiometric flame temperature⁸ (from Sawyer et al., 1973)

where O atom equilibrium is assumed. Using the above result, the chemical time for NO formation is taken as

$$\tau_{\text{no}} = 10^{-12} e^{135 \text{ kcal/mole}/RT_f} \text{ msec} \quad 2-15$$

where the magnitude of the pre-exponential factor has been chosen to yield ratios of characteristic times of order unity (Section IV). Thus the characteristic time for NO formation utilized in the present research is dependent only upon the stoichiometric flame temperature. However it should be noted that this maximum temperature within the combustor is dependent upon both the combustor inlet air temperature and the combustor pressure. Consequently, τ_{no} will decrease (i.e. NO formation rate increases) with either increasing inlet air temperature or increasing combustor pressure.

Following a similar approach to that used above for determining τ_{no} , disc combustor CO emissions from the present experiments and from the experiments of Dodds et al. (1975) on the same burner were normalized to a constant mixing (quenching) time and fitted to the expression

$$\ln \frac{\text{COEI}}{\text{constant}} = 2200 \text{ cal/mole}/RT_i \quad 2-16$$

where T_i is the combustor inlet air temperature and ranged from 500 to 800°K. This correlating expression for CO emissions was based on the inlet air temperature since the CO emissions originate primarily along the outer portion of the shear layer flame due to quenching of the oxidation reactions by the colder inlet air flow. The CO kinetic time scale is then taken as

$$\tau_{\text{co}} = e^{2200 \text{ cal/mole}/RT_i} \text{ msec} \quad 2-17$$

where the pre-exponential factor is assumed equal to one msec. From Equation 2-17, τ_{co} is seen to decrease (i.e., the rate of CO burnup increases) with increasing inlet air temperature. Alternate expressions

for τ_{CO} based on the burned gas temperature, the average of the inlet air and burned gas temperatures, and the average of the stoichiometric flame and burned gas temperatures were also developed and correspond to activation energies of 7700, 4600, and 41200 cal/mole, respectively.

D. Summary

The characteristic times developed and utilized in the present investigation to analyze combustor pollutant emissions are listed in Table 2-3 for convenience and are somewhat different than the times identified by Mellor (1976) (Table 2-1). The first time listed in Table 2-3 is the fuel droplet lifetime and is essentially unchanged from the earlier work of Mellor (1976) except that only fuel droplet evaporation is considered. In practical, high intensity combustors, the likelihood of individual fuel droplet combustion is remote, and hence this process has been removed from further consideration in the present study. Also the second time listed by Mellor (1976), τ_{fi} , has not been deemed of importance in the present experiments since here only simplex fuel nozzles were employed for fuel atomization.

The third time identified by Mellor (1976) (Table 2-1), τ_{sl} , has been further quantified. The present results show that CO and NO are produced in different regions of the shear layer flame structure and that these regions do have different mixing time scale behavior. Consequently, two shear layer mixing times are required and are the second and third entries of Table 2-3.

Mellor (1976) identified one chemical time, τ_{hc} , to characterize the complete combustion of the fuel to CO_2 . Since CO emissions are of interest in the present work, a two step fuel oxidation process has been considered. In the first step, the hydrocarbon fuel is rapidly oxidized to form CO. The CO is then oxidized by a much slower process to form CO_2 . This latter oxidation reaction has been identified in the present work and is given the symbol τ_{CO} . The NO formation time, τ_{no} , utilized here characterizes the same process as the NO time scale of Mellor (1976).

In a conventional gas turbine combustor, the flame structure is strongly dependent upon the rate of gaseous fuel addition from droplet

Table 2-3
Characteristic Times Controlling Burner Emissions

<u>Time</u>	<u>Symbol</u>	<u>Physical or Chemical Process</u>
Fuel droplet lifetime	τ_{eb}	Droplet evaporation
Eddy dissipation time in the region of the shear layer near the rim of the disc	$\tau_{sl,co}$	Large scale turbulent mixing between colder burner inlet air and hotter partially oxidized combustion products
Eddy dissipation time in the downstream regions of the shear layer near the centerline of the burner	$\tau_{sl,no}$	Large scale turbulent mixing between lower temperature free stream flow and high temperature combustion products
CO burnup time	τ_{co}	Homogeneous kinetics of CO oxidation
NO formation time	τ_{no}	Homogeneous kinetics of NO formation

evaporation (characterized by τ_{eb}) and the subsequent mixing of the fuel vapor with fresh oxidizer to form the shear layer flame zone. The turbulent mixing processes in the upstream portion of the shear layer are characterized by $\tau_{sl,co}$ and those in the downstream portion by $\tau_{sl,no}$. The two chemical times, τ_{co} and τ_{no} , do not influence the basic flame structure but rather are dependent upon it.

The methods of calculating the characteristic times discussed above should be thought of as order of magnitude estimates since each takes on a range of values in an actual burner. However, these times do identify the key physical processes affecting combustor emissions and their use in both burner design and emissions data analysis is demonstrated in Sections IV and V.

III. DESIGN AND OPERATION OF THE EXPERIMENTAL COMBUSTION FACILITY

The experimental combustion facility utilized in this research was conceived and has evolved as a system capable of simulation of the combustion process within full-scale gas turbine can combustors. The basic hardware has been used in the past for examination of real combustors and details of the system can be found in descriptions by Mellor et al. (1972), Tuttle et al. (1973), Shisler et al. (1974), Tuttle et al. (1975), and Dodds et al. (1975).

A. Facility Design and Disc Combustor

Compressed air, for burner operation, is supplied from a blow-down type storage system with a volume of approximately 85 m^3 . Maximum system pressure of 150 atm permits an air flow rate of about 3 kg/sec for 40 min; smaller air flow rates can be maintained for correspondingly greater times. For short periods of time air flow rates of approximately 4.5 kg/sec are possible. Air flow rate is monitored with a differential pressure, orifice type flowmeter and is controlled by a pneumatically operated air throttle valve.

Shown schematically in Fig. 3-1, the combustion facility burns commercial grade liquid propane fuel in a slave burner (a standard Allison T-56 combustor with a duplex pressure atomizer) to supply heated, vitiated inlet air to the test section. Fuel flow rates to both burners are controlled via hand operated valves and are monitored with turbine flowmeters. Test section pressure is controlled by a remotely operated back pressure valve. Consisting of a movable, conical center body, the valve determines the available flow exit area. Before being released to the atmosphere, the exhaust flow is quieted by a commercial gas turbine exhaust silencer.

At the entrance to the test section, the temperature of the incoming gas is estimated by means of a temperature rake mounted across

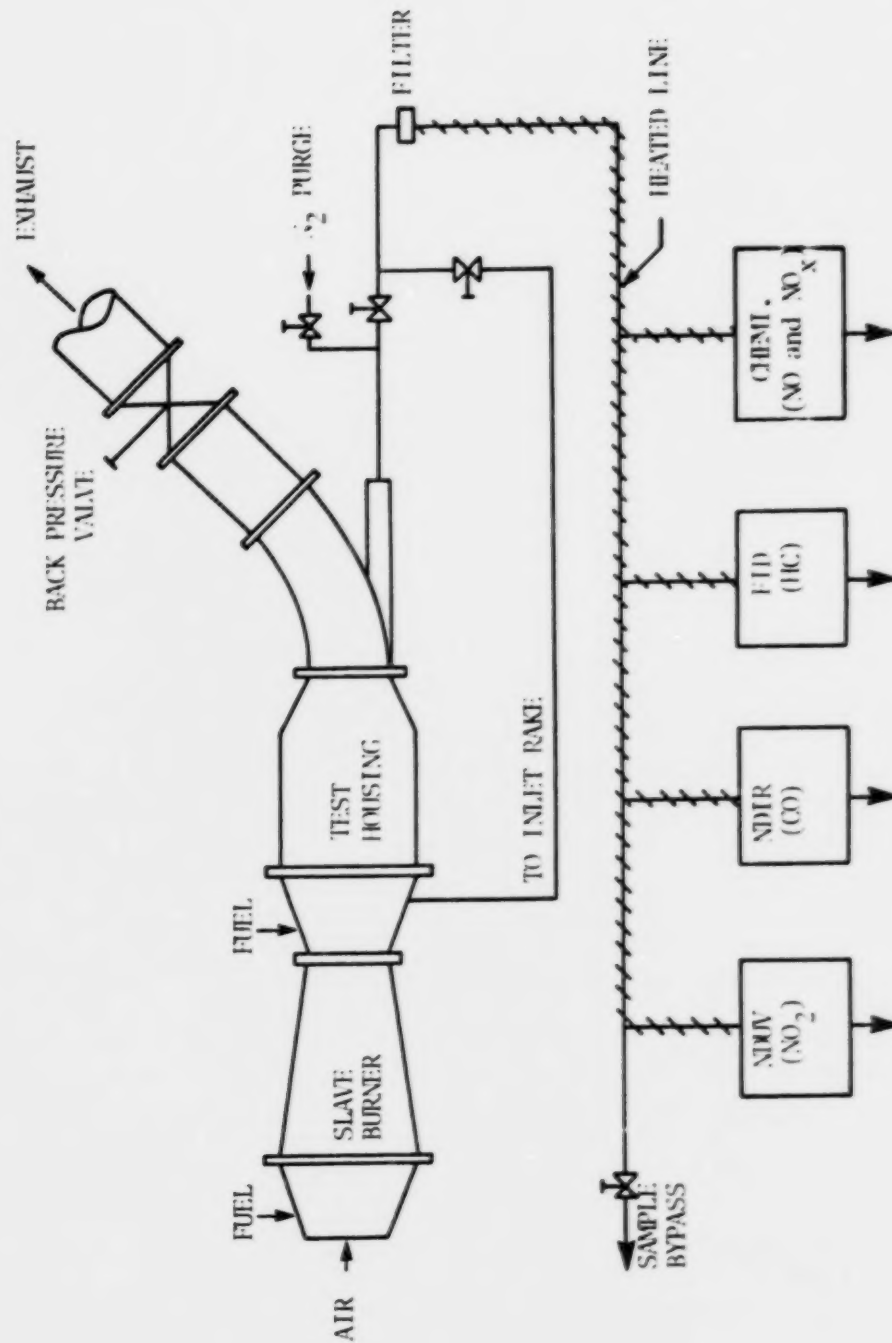


Figure 3-1. Schematic of the experimental apparatus

the duct. Eight chromel-alumel thermocouple beads are positioned along a stainless steel "V" shaped bar such that they will provide an area averaged temperature. Protruding through the point of the "V", the thermocouple wires are insulated from the bar by ceramic Al_2O_3 . Inlet air temperature is adjusted by controlling the fuel flow to the heating combustor.

Just downstream of the thermocouple rake, samples of the vitiated inlet air to the test section are extracted and sent to the gas analysis system via an uncooled gas sampling rake. Constructed of stainless steel tubing in the shape of a "cross", the rake has a total of sixteen collection holes, spaced so as to give an area-averaged sample.

Area-averaged gas samples at the exit plane of the test burner are gathered through the use of a water-cooled stainless steel rake (described by Shisler et al., 1974). This rake is mounted through the window in the elbow section (Fig. 3-1) and has sixteen sampling ports, four located along each of the four spokes which extend from a central hub.

The above described basic facility is capable, with the exception of vitiated inlet air, of effectively producing in the test section conditions representative of gas turbine can type combustor operation. Inlet temperatures have been varied from 273 to 950°K, the upper value being typical of regenerative combustor operation. Test chamber operating pressures from slightly greater than one atm to approximately 10 atm are routinely attainable. It should be emphasized at this point that, over the range of the system, it is possible to vary any one of the basic combustor operating parameters (inlet temperature, pressure, air flow rate or overall equivalence ratio) independently of the others. In addition, either liquid propane or Jet-A fuel can be supplied to the test burner.

For the present experiments, the test section housing was made of stainless steel pipe of inner diameter 14.63 cm. Depicted schematically in Fig. 3-2, the test section had five 3.45 cm diameter quartz windows mounted 6.35 cm apart to allow visual access to the test section interior. Figure 3-2 also shows the downstream reducing section of the housing

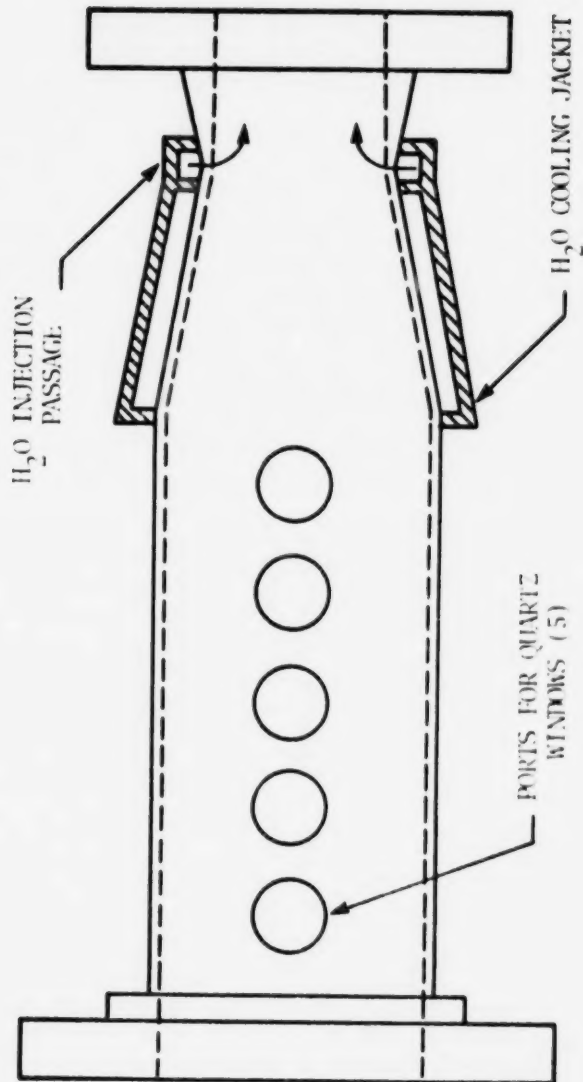


Figure 3-2. Test section housing schematic

which was kept at an acceptable temperature by circulating water through the cooling jacket. In addition, high pressure water was injected into the gas flow to protect the downstream equipment from the high temperature of the combustor exhaust gases.

In all experiments, a disc mounted flush with the edge of the first window was used as a flameholder. Shown in Fig. 3-3, the disc was mounted to the housing by thin support struts oriented to the air stream in a manner to minimize flow disturbances. The disc has a liquid fuel pressure atomizer mounted at its center such that the hollow cone fuel spray originates from the plane of the disc. All mounting arrangements were such that both the disc and atomizer could be easily changed. It was found that both the disc stabilized flame and the heating combustor could be ignited conveniently using CH_4/O_2 torch type igniters located approximately 4 cm downstream of the fuel nozzles.

Single point probe experiments for the above described test burner were previously reported (Tuttle et al., 1976). From the results, the authors were able to formulate a physical model of the disc stabilized flame structure. Shown in Fig. 3-4, the model consists of three regions, central recirculation zone, shear layer flame zone, and a free stream flame zone. Within the recirculation zone, the hollow cone fuel spray was found to penetrate nearly to the shear layer separating the free stream and the recirculation zone. Smaller fuel droplets and fuel vapor were found to flow toward the disc in the backflow region of the recirculation zone and move radially outward along the disc toward the point of the flow separation. At the rim of the disc where mixing between the fuel-rich recirculated gases and the oxygen-rich free stream begins, Tuttle et al. (1976) noted a hollow, but intense, shear layer flame zone begins and extends downstream along the boundary between the free stream and the recirculation zone. In addition, the authors noted cases where a portion of the fuel spray (largest droplets) actually reached and extended through the shear layer and into the free stream. In these latter cases, the flame structure exhibited a free stream flame zone which followed the trajectory of the fuel spray (Fig. 3-4).

The dominant regions of pollutant emissions generation from the disc stabilized flame were also reported by Tuttle et al. (1976) and are

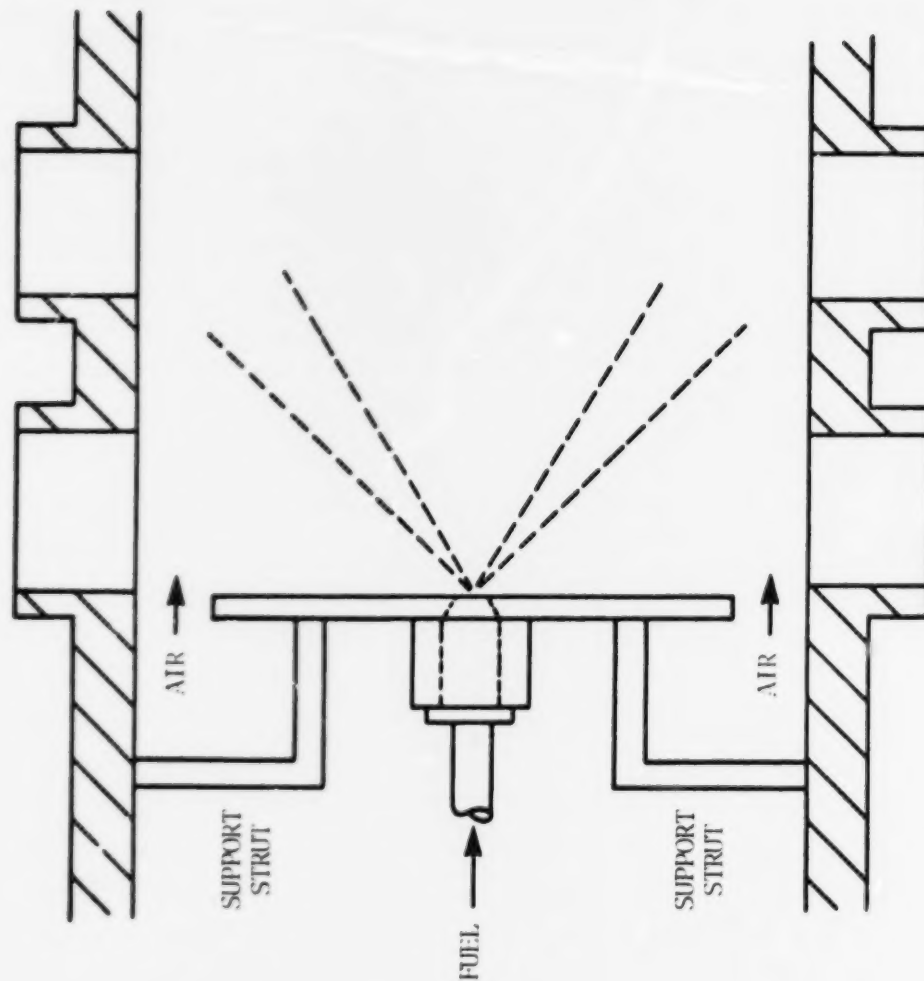


Figure 3-3. Schematic of disc mounting arrangement

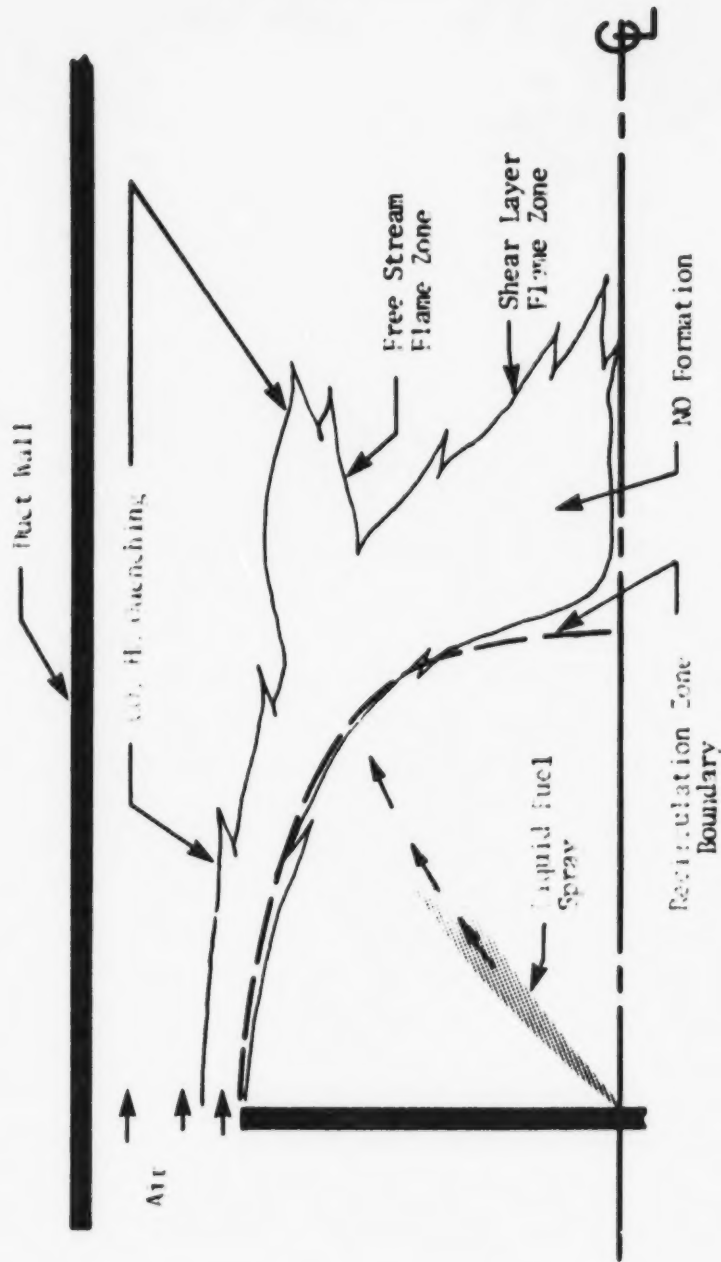


Figure 3-4. Burner and flame schematic

summarized in Fig. 3-4. CO and HC emissions were found to originate both from the outer shear layer region immediately downstream of the disc edge and as a result of large droplets penetrating through the shear layer into the colder free stream. NO formation was found to be most significant in the near stoichiometric, lower velocity portion of the shear layer near the centerline of the burner. The described flow model of the disc stabilized flame with its inherent regions of pollutant formation were emphasized in the previous development of the time scale parameters.

B. Gas Analysis System

Gas samples extracted from either the test section or the inlet gas rake (see Fig. 3-1) were channeled, by the use of appropriate valves, to the gas analysis system. From the valves to the instruments, the 8 m of stainless steel sample line were maintained at approximately 425°K by thermostatically controlled resistive heat tapes after Vaught et al. (1971). The sample flow passed through a glass fiber filter to remove carbon particulate (98% collection efficiency for 0.05 micron particles). The sample stream was analyzed on a continuous basis for concentrations of carbon monoxide, unburned hydrocarbons, nitric oxide and nitrogen dioxide. Concentrations of carbon monoxide and unburned hydrocarbons were measured by a nondispersive infrared analyzer (Beckman model 315A) and a flame ionization detector (Beckman model 402), respectively. A chemiluminescent NO analyzer (similar to that of Fontijn et al., 1970) equipped with an NO₂ stainless steel converter allowed alternate monitoring of NO and total NO_x (NO + NO₂) levels. NO₂ concentrations were measured via a nondispersive ultraviolet analyzer (Beckman model 255B).

Each of the gas analysis instruments produced an output signal in the form of a meter deflection which was visually observed and manually recorded. In addition, a complete, permanent record of the output signal was made by strip chart recorders connected to each instrument. Similar records were obtained for the voltage readings of the temperature monitoring thermocouples.

C. Exhaust Plane Studies

Exit plane emissions data for independent variations in overall equivalence ratio, air flow rate, combustor pressure and inlet air temperature were collected with the water cooled, area averaging rake located 32.2 cm downstream of the disc. The selected operating conditions for the propane experiments are listed in Table 3-1. Since Jet-A emissions data for inlet air temperature variation were previously reported by Dodds et al. (1975), the final two run conditions listed in Table 3-1 were not included in the Jet-A fuel experiments. The procedure for gathering the exit plane data was, after adjusting the combustion system to the desired operating parameters, to measure the values of the incoming pollutant concentrations via the inlet gas sampling rake and then to measure the emissions of the test burner.

Seven disc flameholders (shown schematically in Fig. 3-5 and representing variation of burner geometry) were employed in the propane fuel experiments. Gas samples were collected for three circular disc diameters (6.99, 8.89, and 11.43 cm) and four toothed discs. The three circular discs represent duct area blockage ratios of 22, 36, and 60% respectively. The toothed disc flameholders were employed to further vary the turbulence properties of the outer shear layer region of the flame structure near the rim of the disc. Two toothed discs (8 teeth and 23 teeth) were designed to have the same duct area blockage ratio as the 8.89 cm diameter circular disc; in addition, two similar discs (13 teeth and 29 teeth) had the same duct area blockage ratio as the 11.43 cm diameter circular disc. In each case, the teeth had a tooth depth equal to the average tooth width (aspect ratio of unity). For reference, the tooth widths, W , of the 29, 23, 13, and 8 teeth discs are 0.62, 0.61, 1.37 and 1.71 cm respectively. Burning Jet-A fuel, gas samples were gathered only for the 8.89 and 11.43 cm circular discs.

The effect of liquid fuel droplets upon emissions measurements was investigated by repeating the run conditions listed in Table 3-1 for a range of fuel injector sizes. All test flame fuel injectors were simplex pressure atomizers and were obtained from the Delevan Manufacturing Company. Those used each had a nominal spray cone angle of 90° and were

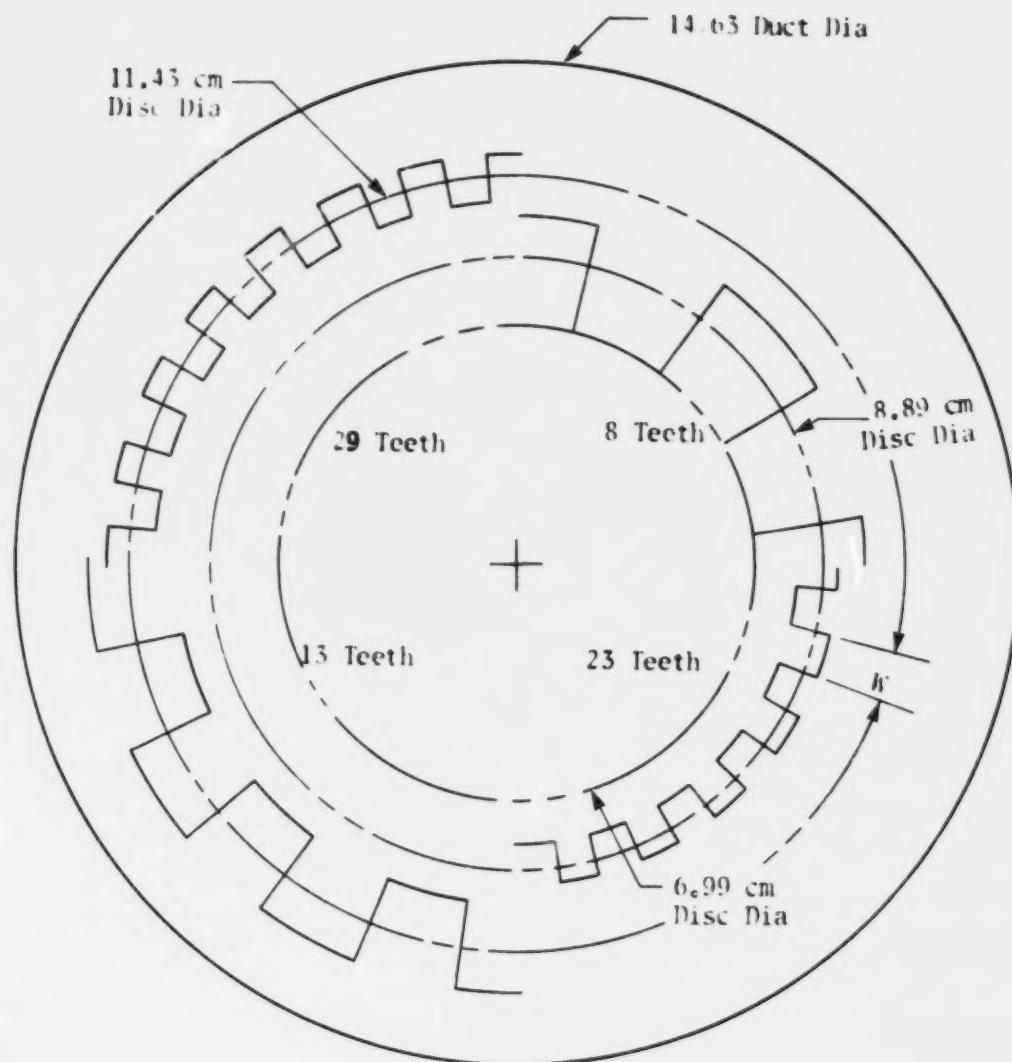


Figure 3-5. Disc flame stabilizers

Table 3-1
Burner Operating Points

Overall Equivalence Ratio	Air Flow Rate (kg/sec)	Combustor Pressure (atm, abs)	Inlet Air Temperature (°K)
0.2	1.0	2.0	500
0.2	1.0	4.0	500
0.2	1.0	5.0	500
0.2	1.0	8.0	500
0.2	0.75	6.8	500
0.2	1.5	5.0	500
0.1	1.0	5.0	500
0.3	1.0	5.0	500
0.2	1.0	5.0	450
0.2	1.0	5.0	550

rated by the supplier at 10, 16, 24, and 35 gal/hr volumetric flow rate of water at an injection pressure of 125 psid. Hence for a fixed fuel flow rate, the differential pressure drop across the nozzle decreased with increasing nozzle size which in turn according to Equation 2-7 resulted in decreased atomization efficiency. In addition, use of the two widely different fuels, propane and Jet-A, greatly varied the fuel spray atomization.

The variety of emissions data obtained in the combustion facility burner described above will be reported and fully discussed in the following section. These measurements will be used to guide the development of emissions correlating parameters from the earlier discussed characteristic times.

IV. EXPERIMENTAL MEASUREMENTS: RESULTS AND DISCUSSION

The interaction between the aerodynamics of the disc flame stabilizer and the fuel spray to form the flame structure has been discussed in earlier sections. While exhaust plane pollutant emissions measurements are not as helpful as internal flame data in analyzing the flame structure, studies of this type do exhibit the dependence of emissions upon the parameters controlling the combustion process. In particular, emissions data for independent variations of the combustor input parameters will be shown to guide the development of data correlating parameters which are useful over the range of burner operation.

As discussed in Section III, burner exit plane gas samples were gathered via a water-cooled, area-averaging rake for the range of burner run conditions listed in Table 3-1. Following a listing of the calculated magnitudes of the time scales for the experimental run conditions, the obtained data will be analyzed in terms of the characteristic times (or combinations thereof) introduced in Section II.

A. Calculation of Characteristic Times

The characteristic time parameters utilized in the present experimental investigation are listed in Table 2-3. In this sub-section, necessary details and assumptions required for the calculation of these time scales for the range of burner operation are presented.

In order to calculate values of τ_{eb} (Equation 2-2), the magnitude of d_o and β must first be estimated. Assuming an ambient gas temperature of 1000°K (typical vaporization zone temperature; see Tuttle et al., 1976) and taking the necessary physical data for liquid propane from Maxwell (1957), β_o was calculated for this fuel via Equations 2-4 and 2-5. Similar calculations for Jet-A fuel were made with the necessary fuel property data taken from Moses (1975). As a result of the wide variation in components within Jet-A and not wanting to bias the

calculations by a small fraction of high boiling point compounds, the vaporization temperature of this fuel was taken to be the 90% vaporized point. Values of β_0 ranged from 0.022 to 0.023 cm^2/sec for liquid propane and from 0.012 to 0.013 cm^2/sec for Jet-A. These values of the evaporation coefficient were then corrected for forced convection effects using Equation 2-6 and assuming a relative velocity between the droplet and gas flow of 50 m/sec following Mellor (1972). This in turn resulted in values of β ranging from 0.04 to 0.06 cm^2/sec and from 0.02 to 0.04 cm^2/sec for liquid propane and Jet-A fuel, respectively.

As stated earlier, the initial fuel droplet diameter was taken as the Sauter mean diameter of the fuel spray and was calculated for the various run points via Equation 2-7. The fuel nozzle differential injection pressure required for the calculation of τ_{eb} was experimentally determined for the desired fuel flow rate. The SMD of the fuel spray ranged from 30 to 80 microns when liquid propane fuel was injected and from 100 to 200 microns when the burner was fueled with Jet-A.

The calculated values of τ_{eb} for the various experimental run conditions and nozzle sizes are listed in Tables 4-1 and 4-2 for liquid propane fuel and Jet-A fuel, respectively. As can be seen in Table 4-1, the values of τ_{eb} for liquid propane fuel are all less than one millisecond. These rather low values are expected, however, as a consequence of the high volatility of liquid propane. The values of τ_{eb} listed in Table 4-2 for Jet-A fuel are seen to be increased over those for propane by nearly an order of magnitude. The greatly increased evaporation times for Jet-A fuel result primarily from the increased viscosity of Jet-A over propane and hence from Equation 2-7 a correspondingly greater SMD of the fuel spray. Also in Table 4-2, note the increase in τ_{eb} at a fixed run condition when the size of the fuel nozzle is increased. Based on the τ_{eb} calculations of Tables 4-1 and 4-2, the heterogeneous effects within the combustion process are expected to be minimal for the propane experiments and more pronounced with Jet-A fuel.

Values of the velocity of the gas flow through the annulus at the disc location were calculated directly from Equation 2-9. This in turn allowed the calculation of $\tau_{\text{sl,co}}$ for the three circular discs ($L_{\text{co}} = d$) via Equation 2-10. For the toothed discs, the integral length

Table 4-1
Calculated Values of τ_{eb} , Propane Fuel

Run Condition ($T_i = 500^\circ\text{K}$)		τ_{eb} msec	
$\dot{m}$ kg/sec	P atm	ϕ_o	fuel nozzle rating, gal/hr at 125 PSID
			16 24 35
1.0	2.0	0.2	0.4 0.6 0.7
1.0	4.0	0.2	0.4 --- ---
1.0	5.0	0.2	0.3 0.5 0.6
1.0	8.0	0.2	0.3 0.5 0.7
0.75	6.8	0.2	0.4 --- ---
1.5	5.0	0.2	0.3 0.5 0.6
1.0	5.0	0.1	--- --- 0.7
1.0	5.0	0.3	--- --- 0.6

Table 4-2
Calculated Values of τ_{eb} , Jet-A Fuel

Run Condition ($T_i = 500^\circ\text{K}$)		τ_{eb} msec					
$\dot{m}$ kg/sec	P atm	ϕ_o	fuel nozzle rating, gal/hr at 125 PSID				
			<u>10</u>	<u>16</u>	<u>24</u>	<u>35</u>	
1.0	2.0	0.2	---	4.6	5.3	---	
1.0	4.0	0.2	---	---	---	---	
1.0	5.0	0.2	---	5.6	4.5	---	
1.0	8.0	0.2	---	5.2	3.7	---	
0.75	6.8	0.2	1.4	4.4	---	---	
1.5	5.0	0.2	---	2.6	3.8	6.5	
1.0	5.0	0.1	2.3	---	---	---	
1.0	5.0	0.5	---	2.6	3.7	6.3	

scale is given by Equation 2-11 and contains an unspecified proportionality constant k . Using optimization techniques to be discussed later, a value of 0.013 was assigned to k for the calculations. The relevant mixing time for NO formation in the shear layer, $\tau_{sl,no}$, was estimated directly from Equation 2-12 where T_b was taken as the computed bulk temperature of the burned gas at the exhaust plane.

The estimated values of $\tau_{sl,no}$ and $\tau_{sl,co}$ for the various run conditions and disc geometries are listed in Tables 4-3 and 4-4 for the circular and toothed discs, respectively. Note that $\tau_{sl,no}$ and $\tau_{sl,co}$ are both varied by at least a factor of six. The shortest times represent run points where reaction quenching rates are high whereas for long values of these times, there is sufficient time for CO burnup and significant NO formation.

The chemical time for NO formation, τ_{no} , can be calculated via Equation 2-15 once the stoichiometric flame temperature is determined. Here the stoichiometric flame temperature was calculated for chemical equilibrium with the computational scheme of Gordon and McBride (1971) and does depend on the initial reactant temperatures and the system pressure. The chemical time for CO burnup was obtained from Equation 2-17 which requires only knowledge of the inlet air temperature.

Values of τ_{no} and τ_{co} as well as the stoichiometric flame temperature are listed in Table 4-5 for the various run conditions of interest. In addition, similar calculations are presented which correspond to the operating points of Dodds et al. (1975) whose emissions data will also be included and analyzed in the present work. For the run points contained in Table 4-5, τ_{no} varies by a factor of 4.5 and τ_{co} , which is less sensitive to inlet air temperature, by a factor of 2.3. The smaller characteristic chemical times are representative of run points where CO burnup and NO formation are rapid.

Again it should be emphasized that the values of the characteristic times given in Tables 4-1 through 4-5 are considered only order of magnitude estimates of the actual values. Uncertainties exist in the initial fuel droplet diameters, turbulent length and velocity scales, and the pre-exponential factors of the chemical times. However these times do reflect the variation of burner operating parameters (air flow,

Table 4-3

Values of $\tau_{sl,no}$ and $\tau_{sl,co}$ (Circular Discs)

Run Condition ($T_i = 500^\circ K$)		6.99 cm disc diameter		8.89 cm disc diameter		11.43 cm disc diameter	
$\dot{m}$ kg/sec	P atm	$\tau_{sl,no}$ msec	$\tau_{sl,co}$ msec	$\tau_{sl,no}$ msec	$\tau_{sl,co}$ msec	$\tau_{sl,no}$ msec	$\tau_{sl,co}$ msec
1.0	2.0	2.2	2.6	1.4	1.4	1.8	1.1
1.0	4.0	2.8	3.2	3.5	3.4	4.5	2.7
1.0	5.0	4.4	5.2	5.7	5.6	7.2	4.4
0.75	6.8	5.0	5.8	6.4	5.5	8.1	4.9
1.5	5.0	-----	-----	2.3	2.3	3.0	1.8
1.0	5.0	-----	-----	4.2	3.4	5.3	2.7
1.0	5.0	-----	-----	3.0	3.4	3.8	2.7

Table 4-4

Values of $\tau_{sl,co}$ (Toothed Discs)

Run Condition ($T_i = 500^\circ\text{K}$)			$\tau_{sl,co}$ msec			
$\dot{m}$ kg/sec	P atm	ϕ_o	8.89 cm disc diameter, 8 teeth	8.89 cm disc diameter, 23 teeth	11.43 cm disc diameter, 13 teeth	11.43 cm disc diameter, 29 teeth
1.0	2.0	0.2	1.3	1.2	1.0	0.9
1.0	5.0	0.2	3.2	2.9	2.4	2.2
1.0	8.0	0.2	5.3	4.7	4.0	3.6
.75	6.8	0.2	5.2	4.6	4.4	4.0
1.5	5.0	0.2	2.2	1.9	1.6	1.5

Table 4-5
Values of τ_{no} and τ_{co}

T_i °K	P atm	T_f °K	τ_{no} msec	τ_{co} msec
500	2.0	2378	2.6	9.2
500	4.0	2397	2.1	9.2
500	5.0	2406	1.9	9.2
500	6.8	2413	1.7	9.2
500	8.0	2418	1.6	9.2
600	3.0	2429	1.4	6.3
700	2.4	2463	0.97	4.9
700	3.0	2469	0.91	4.9
700	5.0	2492	0.70	4.9
700	6.0	2497	0.66	4.9
800	3.0	2509	0.58	4.0

pressure, inlet air temperature, geometry, fuel type, etc.) upon exhaust emissions levels. Thus by independently varying the characteristic times, it is possible to ascertain the relative importance of each characterized subprocess within the overall combustion scheme.

B. Correlation of Emissions (Propane Fuel Experiments)

In the analysis of conventional gas turbine combustors contained in Section II, it was concluded that when fuel and oxidant are fed separately, the resulting flame structure resembles a turbulent diffusion flame. The combustion process is thus mixing controlled and combustion occurs only over a narrow range of equivalence ratios centered about unity. This combustion model implies that for fixed inlet air temperature and pressure (i.e. constant τ_{no} and τ_{co}) and negligible fuel droplet effects, NO_x and CO emissions will be determined only by the rate at which the hot combustion products are diluted with colder species.

Consider the formation of NO within a turbulent eddy of high temperature burned gas. Since all combustion occurs near unity equivalence ratio, the rate of NO formation within this fluid element depends primarily upon the inlet air temperature and pressure prior to combustion since these determine the burned gas temperature. Stated otherwise, only those fluid elements containing burned gas of unity equivalence ratio need be considered since only in those eddies will NO production be significant. The functional dependence of the rate of NO formation upon the burned gas temperature is expressed by τ_{no} (Equation 2-15) which is proportional to the reciprocal of the rate constant of Reaction 2-14. Since [O] and $[N_2]$ are present in excess, the reaction rate is taken as zero order.

$$\frac{d[NO]}{dt} / \begin{matrix} \text{burned gas} \\ \text{eddy} \end{matrix} = f(T_i, P) \sim \frac{1}{\tau_{no}} \quad 4-1$$

NO formation continues in the high temperature eddies at the above rate until the reaction rate is limited by the mixing of colder species into the hot combustion products. Since this mixing process is characterized by $\tau_{sl,no}$, integration of Equation 4-1 yields

$$[NO] / \begin{matrix} \text{burned gas} \\ \text{eddy} \end{matrix} \sim \frac{\tau_{sl,no}}{\tau_{no}} \quad 4-2$$

To determine the total NO generated by the overall combustion process, a sum must be taken over all of the burned gas eddies. However the number of burned gas eddies of interest in the system should be proportional to the amount of fuel supplied to the burner. The total NO emissions are then just proportional to the product of the mass of the fuel supplied and the time ratio.

$$[NO]_{total} \sim (\text{fuel supplied}) \frac{\tau_{sl,no}}{\tau_{no}} \quad 4-3$$

On dividing both sides of Equation 4-3 by the mass of the fuel supplied to the burner, the NO emissions index, NOEI, is formed.

$$\text{NOEI} \sim \frac{\tau_{sl, \text{no}}}{\tau_{\text{no}}} \frac{\text{g pollutant}}{\text{kg fuel}} \quad 4-4$$

Thus for a diffusion flame burner operating at a fixed pressure and inlet air temperature (fixed τ_{no}) and containing negligible fuel droplet effects, Equation 4-4 predicts that NOEI should remain constant for a fixed mixing time as the fuel flow rate is varied.

Similarly, the rate of CO oxidation with a turbulent eddy of partially oxidized burned gas will be considered. The rate of CO burnup (destruction) within this fluid element depends primarily upon its gas temperature (which in the present work has been characterized by the inlet air temperature) and the concentration of CO.

$$\frac{d[\text{CO}]}{dt} = -f(T_i) [\text{CO}] \sim -\frac{[\text{CO}]}{\tau_{\text{co}}} \quad 4-5$$

where the functional dependence of the rate of CO disappearance upon the inlet air temperature is expressed by τ_{co} (Equation 2-17). CO oxidation continues within the fluid element until the reaction rate is limited by the mixing of colder species into the partially oxidized combustion products. Since this mixing process was characterized in Section II by $\tau_{sl, \text{co}}$, integration of Equation 4-5 yields

$$\frac{[\text{CO}]}{[\text{CO}]_0} \sim \exp(-k \tau_{sl, \text{co}}/\tau_{\text{co}}) \quad 4-6$$

where the constant k has been introduced into the right hand expression because of the non-linear dependence of the CO concentration upon $\tau_{sl, \text{co}}/\tau_{\text{co}}$. In particular without the introduction of k into Equation 4-6, the level of CO emissions predicted would depend upon the arbitrarily chosen value of the pre-exponential factor of Equation 2-17 which determines the range of the ratio $\tau_{sl, \text{co}}/\tau_{\text{co}}$. The left hand side of Equation 4-6 represents the fraction of the initial CO concentration remaining in a fluid element after a time $\tau_{sl, \text{co}}$. However since the initial CO concentration in the gas eddy depends upon the fuel flow rate, the left hand side of Equation 4-6 can be taken proportional to COEI.

$$\text{COEI} \sim \exp(-k \tau_{sl,co}/\tau_{co})$$

4-7

The value of k in the above equation cannot be evaluated a priori but must be obtained empirically. Considering the CO emissions from the burner fueled with propane, the best correlation of the data was obtained for a value of k of 9.45. For this value of k and the values of $\tau_{sl,co}$ and τ_{co} utilized in the present study, the exponential function can be approximated closely by a power series of $\tau_{co}/\tau_{sl,co}$.

$$\begin{aligned} \exp(-9.45 \tau_{sl,co}/\tau_{co}) &\approx a_0 + a_1 \frac{\tau_{co}}{\tau_{sl,co}} \\ &+ a_2 \left(\frac{\tau_{co}}{\tau_{sl,co}} \right)^2 + \dots \end{aligned} \quad 4-8$$

where the coefficients $a_0, a_1, a_2 \dots$ were determined by standard numerical techniques.* For the range of values of the argument of the exponential function utilized in the present experimental study, analysis showed that neglect of the squared and higher order terms resulted in an error of less than one percent; consequently, a two term (linear) approximation of the exponential function will be used to correlate CO emissions. In addition to the inherent simplification of a linear correlating parameter, the success of the linear approximation will be insensitive to the value of the pre-exponential factor used in the calculation of τ_{co} . Thus the correlating parameter analogous to Equation 4-4 is taken as

$$\text{COEI} \sim \frac{\tau_{co}}{\tau_{sl,co}} \quad \frac{\text{g pollutant}}{\text{kg fuel}} \quad 4-9$$

*The coefficients in the expansion were determined by minimizing the square of the difference between the exponential function and the approximating series over the range of values of the argument of interest. For a two term expansion, the residual, Δ , is given by

$$\Delta = \int_a^b [\exp(-kx) - a_0 - a_1/x]^2 dx$$

where a and b are the minimum and maximum values of x respectively for the range of x of interest. The coefficients a_0 and a_1 are then determined from the two simultaneous equations:

$$\frac{\partial \Delta}{\partial a_0} = 0 \quad \text{and} \quad \frac{\partial \Delta}{\partial a_1} = 0.$$

In order to experimentally examine the validity of Equations 4-4 and 4-9, measurements of NO_x and CO were gathered for overall equivalence ratios of 0.1, 0.2, and 0.3 at constant values of air flow rate, combustor pressure, and inlet air temperature (i.e. constant τ_{no} , τ_{co} , and $\tau_{\text{sl,co}}$). Later results will show that the effect of liquid fuel droplets upon the flame structure was minimal when the burner was fueled with propane. In presenting the results, the dependence of $\tau_{\text{sl,no}}$ upon overall equivalence ratio (exhaust gas temperature) will be included through division of the NO_xEI values by the appropriate values of $\tau_{\text{sl,no}}$ taken from Table 4-3. Net (test burner measurement less heating combustor measurement) NO_x values* are plotted in Fig. 4-1. The use of net NO_x emissions is necessary since once formed, NO_x is rather stable and emissions from the upstream combustor will also be measured at the exhaust of the test section. Although Fig. 4-1 does show a decrease in NO_xEI with increasing overall equivalence ratio over the three fold change in fuel flow rate, the reduction is small compared to the predictions of chemical kinetics for similar changes in the local equivalence ratio. Thus the basic combustion regime (i.e., near stoichiometric burning) must remain fixed over this extended range of overall equivalence ratio.

The rather constant CO emissions displayed in Fig. 4-1 over the range of overall equivalence ratio variation also suggest the processes leading to CO emissions are unchanged as the fuel flow rate varies. Net CO emissions have not been used since the CO emissions of the heating combustor were typically less than 5% of those measured at the exhaust of the test burner. Note however that if heterogeneous effects are important within the flame structure, COEI would not be constant since the SMD of the fuel spray does decrease with increasing fuel flow.

*The burner emissions of NO (nitric oxide only) and NO_x (total oxides of nitrogen, $\text{NO} + \text{NO}_2$) have been used somewhat interchangeably in the previous discussions. Although it is generally accepted that NO formation determines the level of NO_x emissions from a combustor, the nature of the NO_x emissions remains a topic of much current discussion (see Tuttle et al., 1974). The fact that NO is converted to NO_2 in the atmosphere forms the impetus for reporting all NO_x emissions as NO_2 .

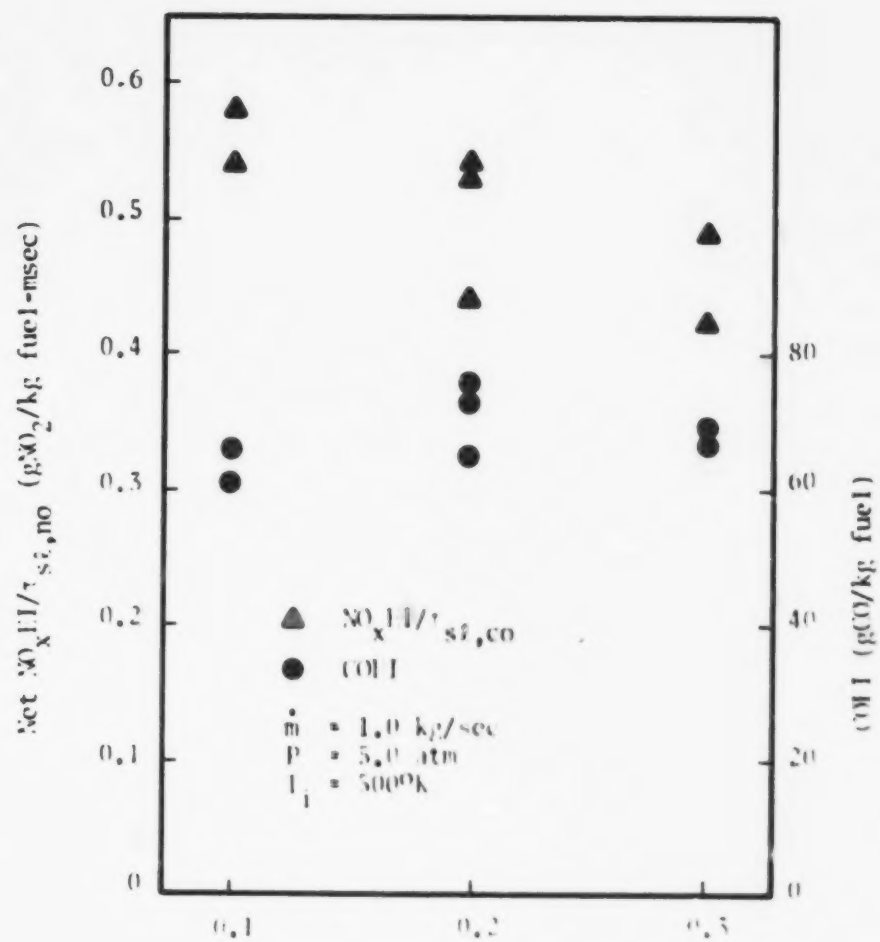


Figure 4-1. Overall equivalence ratio variation (propane fuel)

Similar results for variation of the fuel flow rate at a nearly constant mixing condition and constant τ_{no} and τ_{co} exist in the literature. Tuttle et al. (1973), in studying the emissions characteristics of the Allison J-35 combustor, observed NO_x to decrease from 3.5 to 2.5 gNO₂/kg fuel as the overall equivalence ratio was increased from 0.15 to 0.25. Thus only a 34% decrease in NO_x was found for a 167% increase in the fuel flow rate. If the effective increase in the gas velocity through the combustor due to additional heat release with increasing overall equivalence ratio were accounted for, the NO_x reduction would be even less. In addition for a conventional gas turbine liner, Friswell (1972) observed little change in NO_x over a fourfold change in fuel flow rate while maintaining constant air flow rate and combustor pressure. These two earlier studies together with the present investigation do indeed suggest that the combustion process within a liquid fueled gas turbine combustor occurs only over a narrow range of local equivalence ratios. Hence provided the droplet evaporation time is short, NO_x and CO₂ measurements should correlate according to Equations 4-4 and 4-9, respectively.

The measured net NO_x emissions for the three circular discs, four toothed discs, and the various run points discussed in Section III are plotted in Fig. 4-2 versus $\tau_{st,no}/\tau_{no}$. Although some data scatter is evident, the measurements generally fall closely about the linear relationship suggested on the figure for the various fuel nozzles and flameholders. The goodness of the linear fit is statistically measured by the correlation coefficient (symbol r) where a correlation coefficient of zero represents no correlation and a value of unity represents perfect correlation. In addition, the standard deviation of the NO_x values about the line (symbol σ) is also given on Fig. 4-2. A development and discussion of the statistical concepts utilized in the derivation of the least squares straight line, correlation coefficient, and standard deviation are contained in Appendix A.

The data scatter in Fig. 4-2 would be somewhat reduced if straight lines were drawn through the points corresponding to a particular fuel nozzle. However since a uniform trend between the measured emissions and the size of the nozzles is not observed, the data are in agreement

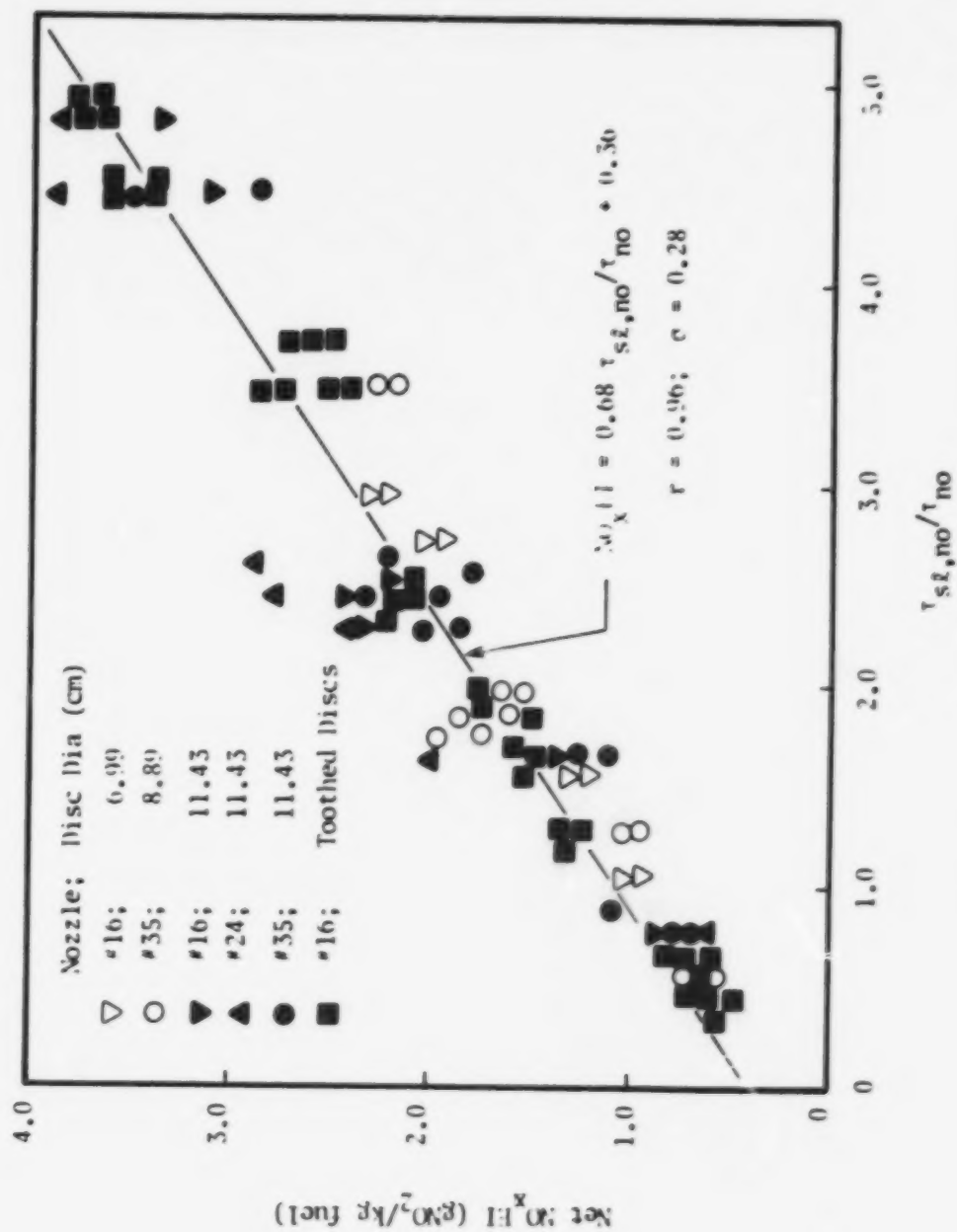


Figure 4-2. NO_x EI versus $\tau_{sl, no} / \tau_{no}$ (propane fuel)

with the τ_{eb} calculations of Table 4-1 which show the droplet lifetimes to be small. Thus due to the high volatility of propane, minimal effect of liquid fuel droplets upon the flame structure was observed experimentally. The data scatter in Fig. 4-2 then must result from run to run measurement irreproducibility rather than an effect of fuel atomization.

Figure 4-2 also shows the NO_x EI data for the two smaller disc flame stabilizers to fall nearly in line with that of the largest disc. Thus credence is given to the selection of the disc diameter as the appropriate integral length scale of the turbulence in the NO producing region of the flame structure. As predicted since the NO formation region is near the burner centerline rather than the disc rim, no difference between the NO_x emissions of the circular discs and toothed discs was observed.

The CO measurements from the propane fuel experiments with the three circular discs are displayed in Fig. 4-3 versus the suggested correlating parameter. Again, good correlation of the data is seen about a linear relationship. In agreement with NO_x EI data, no discernible difference in the CO measurements was observed when the fuel spray atomization was altered through use of three fuel injectors. The rather high value of the correlation coefficient for the straight line through the data extends confidence to the use of the disc diameter and the gas velocity past the rim of the disc as the appropriate turbulent length and velocity scales, respectively.

Since the toothed discs were designed to have the same gas velocity past the rim of the disc as the circular discs, they will also have the same turbulent velocity scales as the circular discs. The experimental data do suggest however that the toothed and circular discs have different turbulent length scales in the CO producing regions of the flame structure. This is demonstrated in Fig. 4-4 where the CO EI measurements are plotted against $\tau_{co}/\tau_{st,co}$ with $\tau_{st,co}$ calculated using the disc diameter as the turbulent length scale (i.e. the values listed in Table 4-3 rather than Table 4-4). Although a general correlation of the CO EI data is seen in Fig. 4-4, the correlating parameter has not successfully collapsed the data from the four discs onto a single linear relationship as overlapping of the data is not seen. In the case

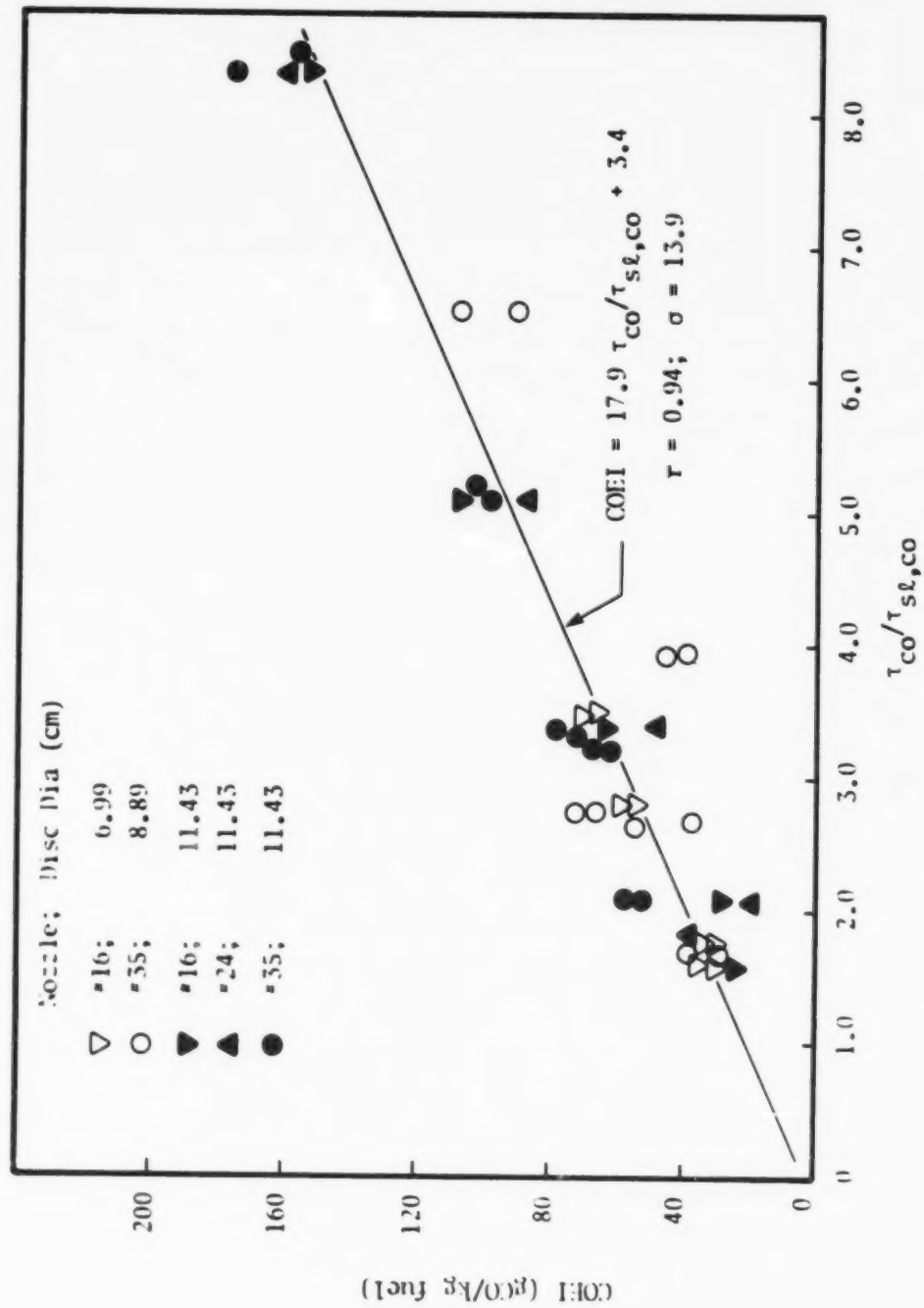


Figure 4-3. COEI versus $\tau_{co}/\tau_{sl,co}$ (circular discs)

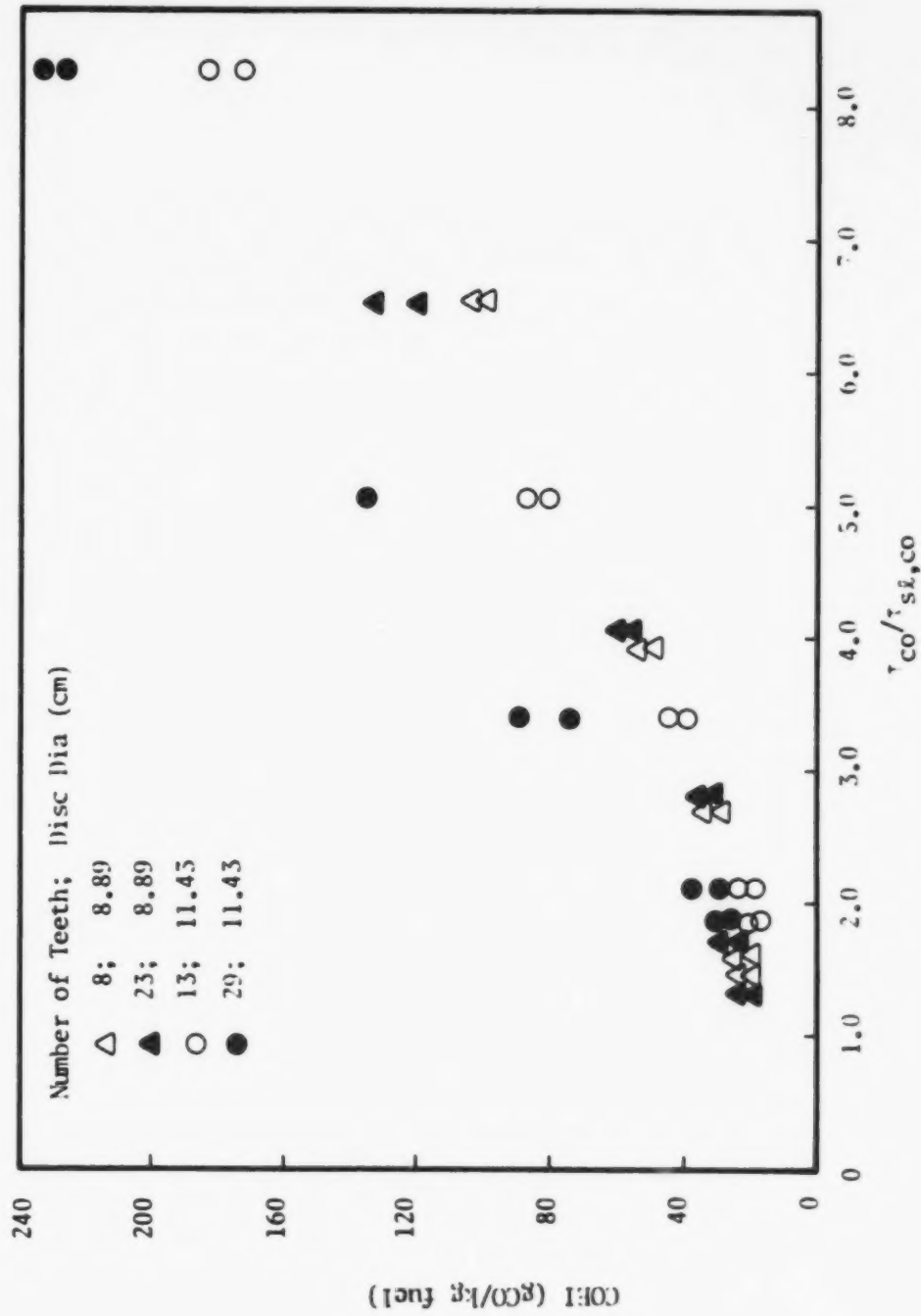


Figure 4-4. COEI versus $\tau_{co}/\tau_{sl,co}$ (toothed discs only, $l_{co} = d$)

of the larger disc diameter, all of the measurements gathered for the 29 teeth disc fall above those of the 13 teeth disc. Similarly for the 8.89 cm discs, the data points for the 23 teeth disc are all greater than the corresponding measurements for the 8 teeth disc. Note however that the difference in measurements for the smaller discs is diminished from the case of the larger discs. In addition, the CO emissions reported in Fig. 4-4 are generally higher than those of the corresponding circular discs. Thus for a toothed disc, the integral length scale of the turbulence in the shear layer immediately downstream of the disc edge must be fundamentally different from the case of a circular disc.

Recall that in Section II for a toothed disc the length scale of the turbulence in the CO producing portion of the flame was suggested to be a weighted average of the reciprocals of the disc diameter and the tooth width (Equation 2-11). This relationship can be rewritten in the form given by Equation 4-10.

$$L_{CO} = \frac{d}{1 + k \frac{d}{W}} \quad 4-10$$

A value of the constant k in the above equation of 0.013 was obtained by optimizing the correlation coefficient of the toothed disc CO data with respect to k . Using this value of k , the $\tau_{sl,CO}$ values (Table 4-4) for the toothed discs were calculated via Equation 4-10.

From the form of Equation 4-10, L_{CO} for the toothed discs can be considered to be a perturbation from the length scale for the circular discs (i.e., $L_{CO} = d$). Note that Equation 4-10 predicts L_{CO} for the toothed discs to decrease (more rapid CO quenching) with decreasing tooth width and that the effective perturbation from the case of a circular disc diminishes with decreasing disc diameter. Both of these effects are advantageous to collapsing the data of Fig. 4-4 onto a single line. In the case of a circular disc, the tooth width can be considered to be the disc circumference such that Equation 4-10 reduces to $L_{CO} = d$.

The CO emissions of the toothed discs along with the circular disc data are replotted in Fig. 4-5 versus the appropriate turbulent length scale. The successful correlation of all the data about the given

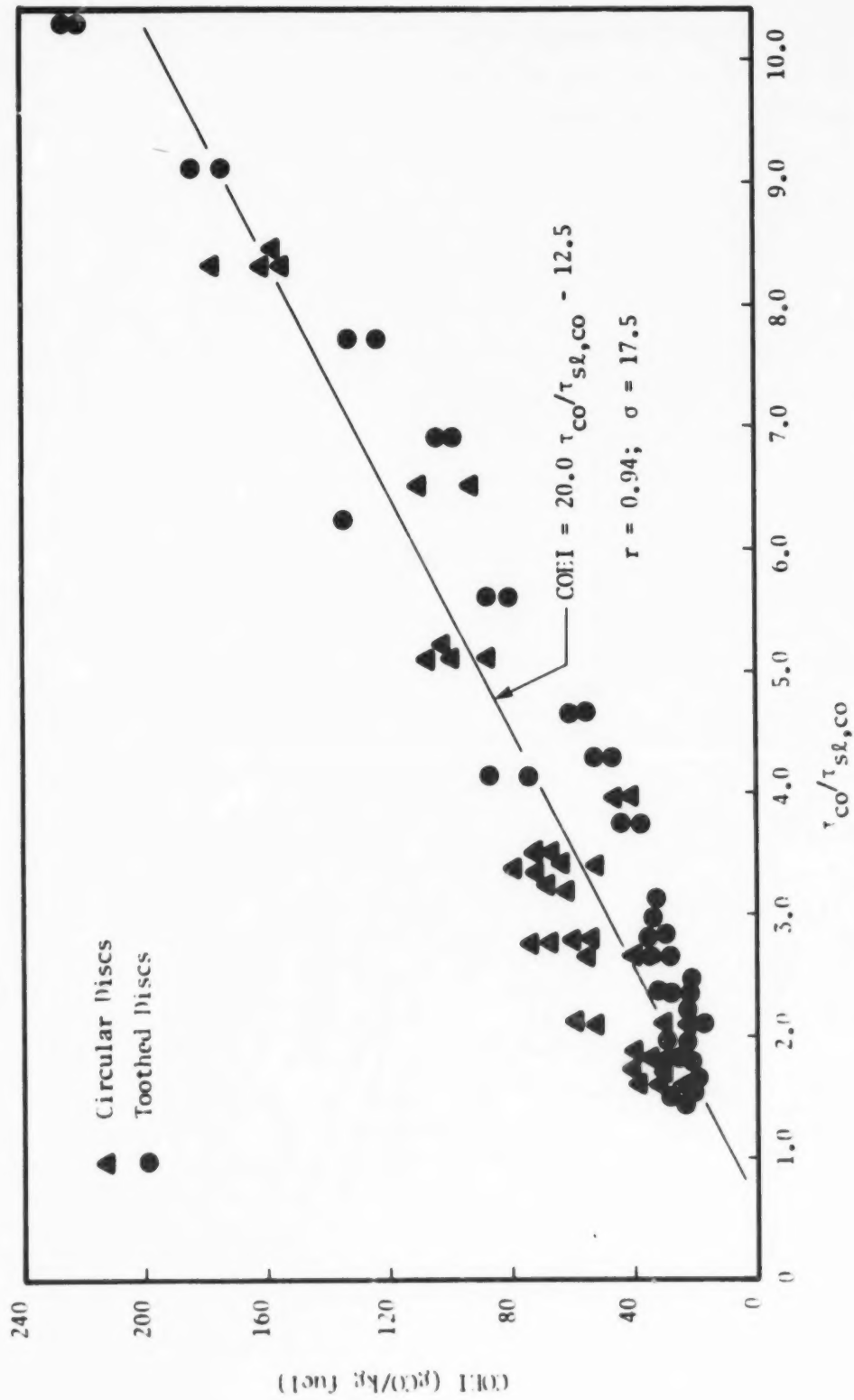


Figure 4-5. COI versus $\tau_{CO}/\tau_{sl,co}$ (propane fuel)

linear relationship is indicated by the relatively high value of the correlation coefficient. The scatter of the toothed disc data displayed in Fig. 4-5 is considerably reduced from that of Fig. 4-4 and in addition, the toothed and circular disc data are overlapped. This improved correlation parameter suggests that for the discs with few teeth (large tooth width), the length scale reduces to that of a circular disc. However, for the 23 and 29 teeth discs, the tooth widths are sufficiently small that the secondary vortices created by flow around the teeth effectively decrease the turbulent length scale.

In the analysis of CO and HC combustor emissions data, it is generally observed that both species exhibit similar trends (Verkamp et al., 1974). From Fig. 4-6 in which HCEI is plotted against COEI, it is seen that this is also true of the disc stabilized flame data. For various fuel injectors and discs, a given level of HCEI corresponds to a unique level of COEI. This is not unexpected, as CO and HC emissions result from the same physical processes. Hence once CO data are well correlated for various run conditions and burner geometries, a similar correlation will exist for HCEI.

Since the dominant CO and NO producing regions of the burner are separate, the possibility exists of reducing both of these pollutants simultaneously by decreasing the correlating parameters associated with each pollutant. Consider the disc burner operating at a fixed inlet air temperature and pressure (i.e., constant τ_{no} and τ_{co}). The burner emissions are then only dependent upon the two mixing times, $\tau_{sl,no}$ and $\tau_{sl,co}$. If in addition a set engine power output level is desired, then the air flow rate and overall equivalence ratio are also fixed. In this constant engine power output case, the duct velocity, V_o , which is dependent upon the burner operating parameters is also necessarily constant and thus the NO_x emissions of the burner are dependent only upon the disc diameter (Equation 2-12). At a set power level, CO emissions from the burner scale with V_a/d where the gas velocity through the annulus at the plane of the disc is related to the duct velocity by a geometric factor. Thus the geometric scaling for CO emissions becomes

$$COEI \sim \frac{1}{d (D^2 - d^2)} \quad 4-11$$

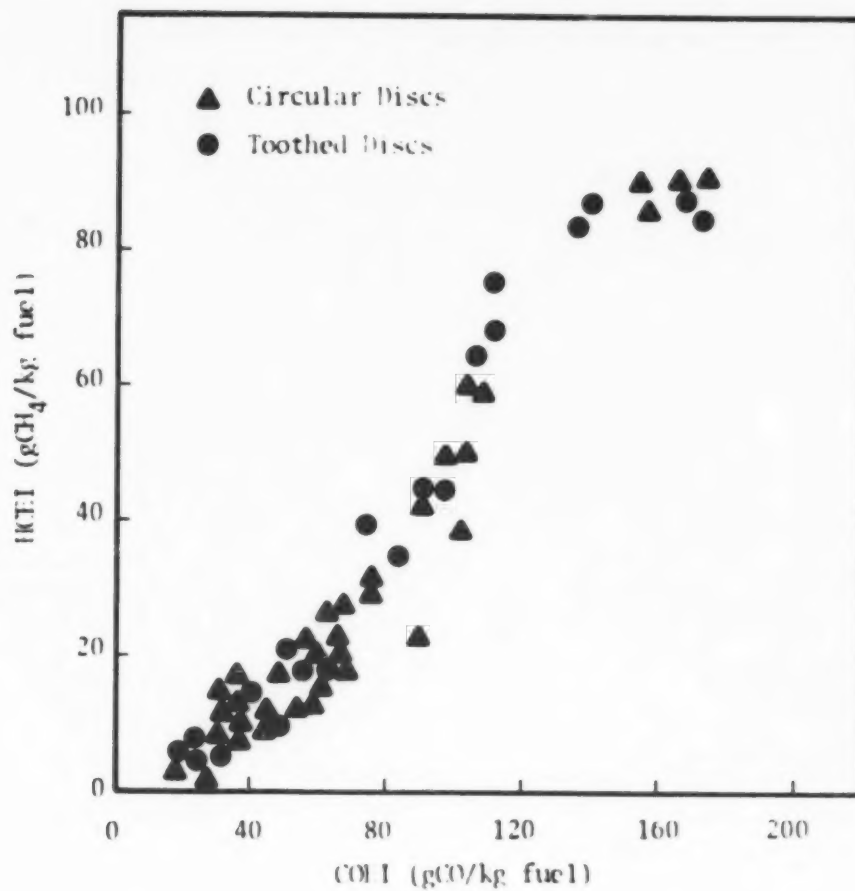
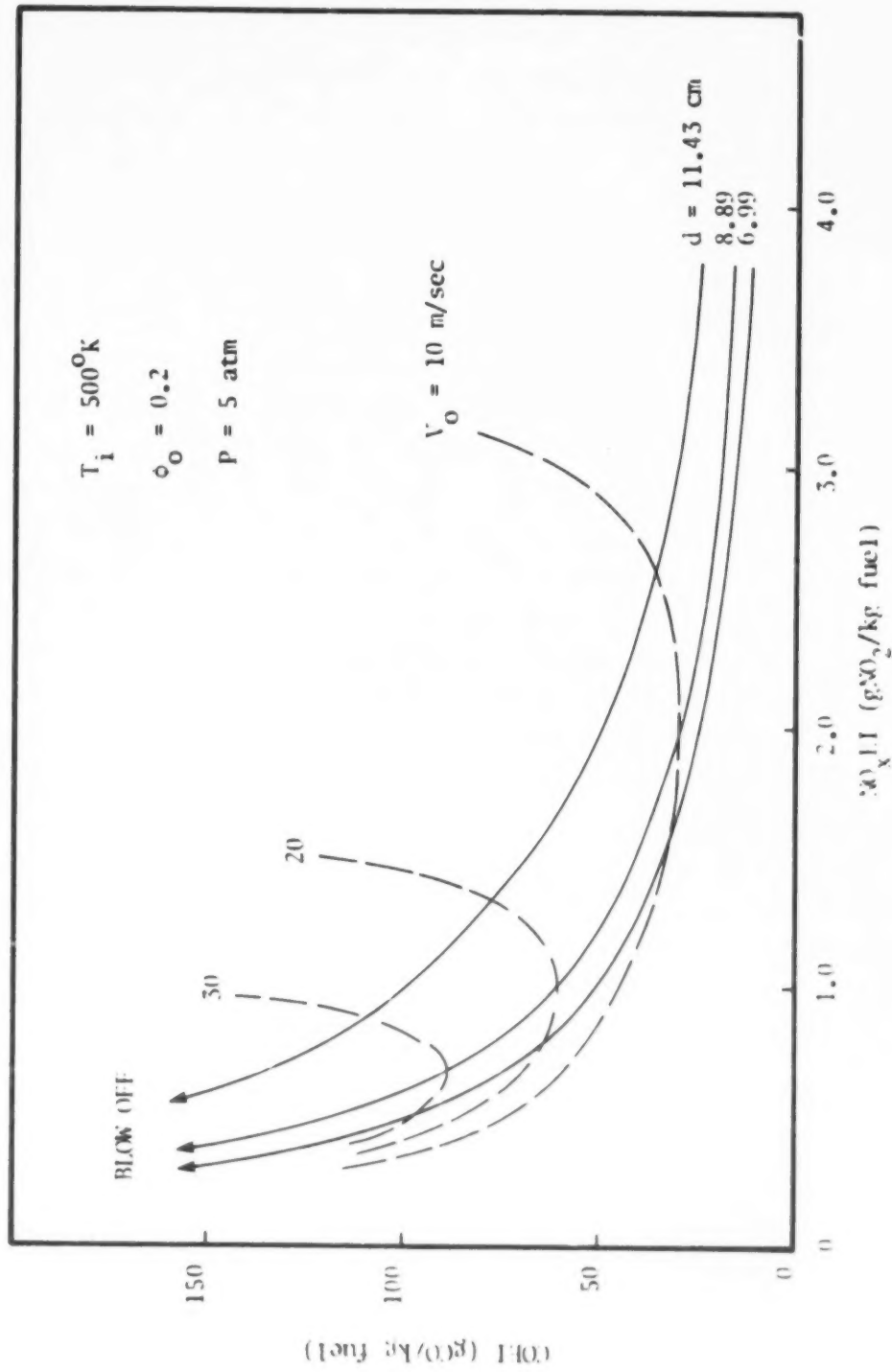


Figure 4-6. HCEI versus COEI (propane fuel)

Hence the scaling relationships predict NO_x emissions from the burner will continually decrease with decreasing disc diameter. However the same is not true for burner CO emissions since Equation 4-11 predicts an optimum disc diameter for minimum CO emissions at a value of d/D of 0.58. For large discs ($d > 0.58D$), the velocity past the disc rim is high and mixing in the CO producing region is too rapid. However if the disc diameter is too small ($d < 0.58D$), the correspondingly small length scale also accelerates the mixing in the outer shear layer.

The effect of disc diameter variation upon both CO and NO_x emissions is demonstrated in Fig. 4-7. For a given disc diameter, the solid lines (hyperbolas) in the figure show a trade-off in CO and NO_x emissions over a range of gas velocities through the duct. Thus if the burner geometry is fixed, run conditions which correspond to low NO_x emissions necessarily correspond to high CO emissions. Note however that as the disc diameter is decreased, the hyperbolas move nearer to the origin of the figure which corresponds to reduced total emissions ($\text{NO}_x + \text{CO}$). There will however be a limit to the extent to which the disc diameter can be reduced since the flame stability of the burner also decreases with decreasing disc diameter because of a corresponding reduction in the size of the recirculation zone behind the disc.

Figure 4-7 also shows the variation in CO and NO_x emissions with changing disc diameter at a fixed gas velocity through the duct. For a constant V_0 , NO_x emissions decrease continually with decreasing disc diameter while CO emissions reflect the optimum disc size (8.50 cm) discussed above. Indeed experimentally CO emissions decreased in going from an 11.43 cm to an 8.89 cm diameter disc and then increased somewhat as the disc diameter was further reduced to 6.99 cm. Note that at high V_0 (high engine power) the disc diameter yielding minimum CO emissions is sharply defined and a significant reduction in emissions of this species results from use of the optimum sized disc. However at low values of V_0 (low engine power), the disc diameter producing minimum CO emissions is less well defined and use of a disc diameter smaller than 8.50 cm will produce a significant reduction in NO_x emissions with only a small increase in CO emissions. Thus Fig. 4-7 demonstrates the useful-

Figure 4-7. COI versus x_{O_2}

ness of variable geometry as a technique to minimize total burner emissions over the range of engine power level.

The exhaust plane measurements of CO and NO_x emissions for the three disc diameters utilized are compared with values of these species measured at the exhaust of existing gas turbine engines in Fig. 4-8. The band of idle to maximum power emissions was obtained from exhaust measurements of a wide range of Detroit Diesel Allison, Pratt and Whitney, and General Electric engines (Verkamp et al., 1974). Also included in Fig. 4-8 is a dashed line representing the predicted emissions characteristics of the disc diameter yielding the minimum CO emissions. As is seen in the figure, the disc stabilized flame has improved emissions characteristics over conventional combustors near idle operation but is comparable at the maximum power mode. Note the decrease in emissions in going from the large to intermediate diameter disc which is near the optimum size. For reference, the Environmental Protection Agency Class T2 aircraft emissions standards are also depicted in Fig. 4-8; the similarity between the emissions data of the present work and those of conventional combustors provides corroborating evidence of the difficulty (Verkamp et al., 1974) of satisfying the EPA standards with diffusion flame combustors.

In summary, the exhaust plane emissions of NO_x and CO for propane fuel have correlated well without including a functional dependence of τ_{eb} in the developed correlating parameters. This experimental finding is in agreement with the relatively low values of τ_{eb} listed in Table 4-1 for liquid propane fuel injection. Thus the use of less volatile Jet-A fuel in a similar set of experiments was necessary in order to determine the effect of liquid fuel droplets upon pollutant emissions.

C. Correlation of Emissions (Jet-A Fuel Experiments)

Exhaust emissions measurements were repeated for the range of operating parameters discussed previously except the fuel was changed to Jet-A, a kerosine type fuel. Hence the fluid mechanic structure of the recirculation zone and shear layer regions will be generally maintained but the fuel droplet lifetime and thus spray penetration will be greatly augmented. The increased heterogeneous effects within the overall

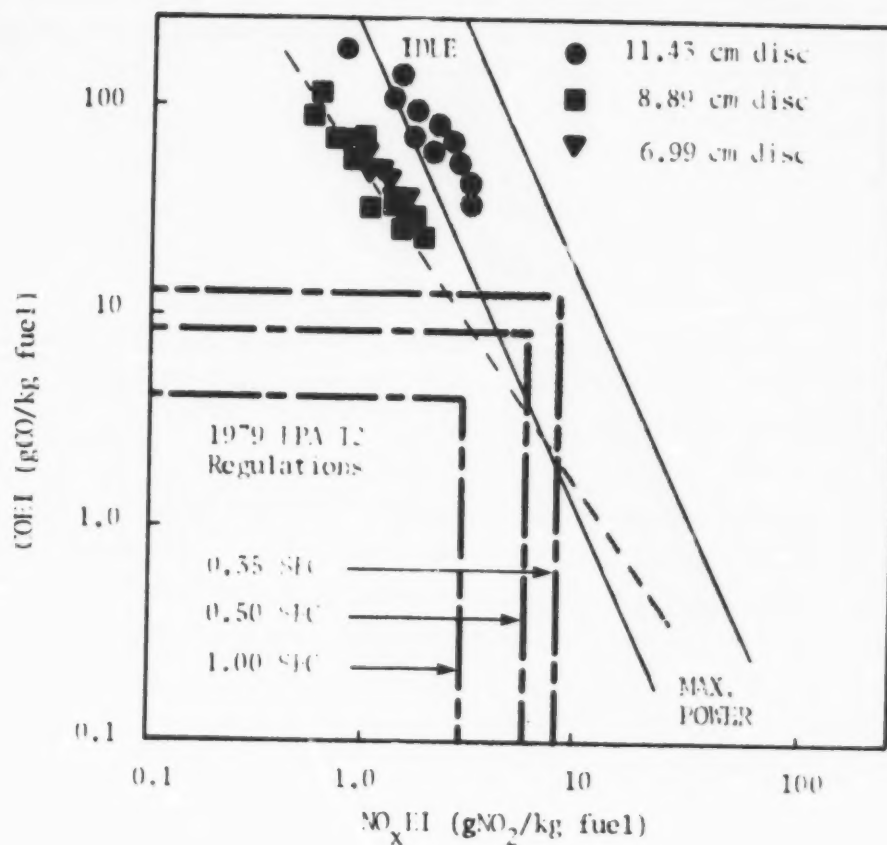


Figure 4-8. Emissions of conventional combustors (from Verkamp et al., 1974). The dashed line represents predicted emissions characteristics for the disc diameter yielding the minimum CO emissions.

combustion process with the use of Jet-A fuel are expected to increase the importance of the free stream flame zone depicted in Fig. 3-4 in relation to the shear layer flame zone.

In order to determine if indeed the measured emissions are dependent upon the fuel spray atomization, NO_x and CO emissions were first plotted against the corresponding parameters used to correlate the propane fuel data. The NO_x data gathered for various nozzle sizes and two circular disc diameters are displayed in Fig. 4-9. The data contained in Fig. 4-9 are seen to correlate reasonably well with $\tau_{sl,no}/\tau_{no}$ which indicates that the mixing and chemical subprocesses continue to dominate the overall combustion process. However it is also apparent that heterogeneous processes are affecting the emissions levels through increased fuel droplet penetration into the free stream at many of the run points.

Since the recirculating flow region behind the smaller disc is decreased in size from that of the larger disc, the effective distance a fuel droplet must travel to pass through the shear layer is diminished for this disc and consequently fuel droplet effects are more pronounced. Notice that for the smaller disc data of Fig. 4-9, the largest nozzle (#24) consistently corresponds to the lowest NO_x emissions whereas the #10 nozzle with its much improved atomization properties yields the highest measurements. The effect of fuel injector upon the measurements with the larger disc is less noticeable, since here the allowable time for a droplet to evaporate prior to passing through the shear layer is increased. Apparently, this increased time for droplet evaporation resulting from the larger recirculation zone is sufficient for nearly all of the fuel droplets produced by the nozzles tested to evaporate before reaching the shear layer. It should also be noted that when a high degree of fuel spray atomization was achieved (#10 nozzle), the data from the two sizes of discs are correlated well by $\tau_{sl,no}/\tau_{no}$ as in the case of propane fuel.

The CO measurements are plotted against $\tau_{co}/\tau_{sl,co}$ in Fig. 4-10 and also show a dependence upon fuel droplet penetration. In general, the measurements corresponding to the smaller disc all lie well above those for the larger disc. When a high degree of atomization was produced,

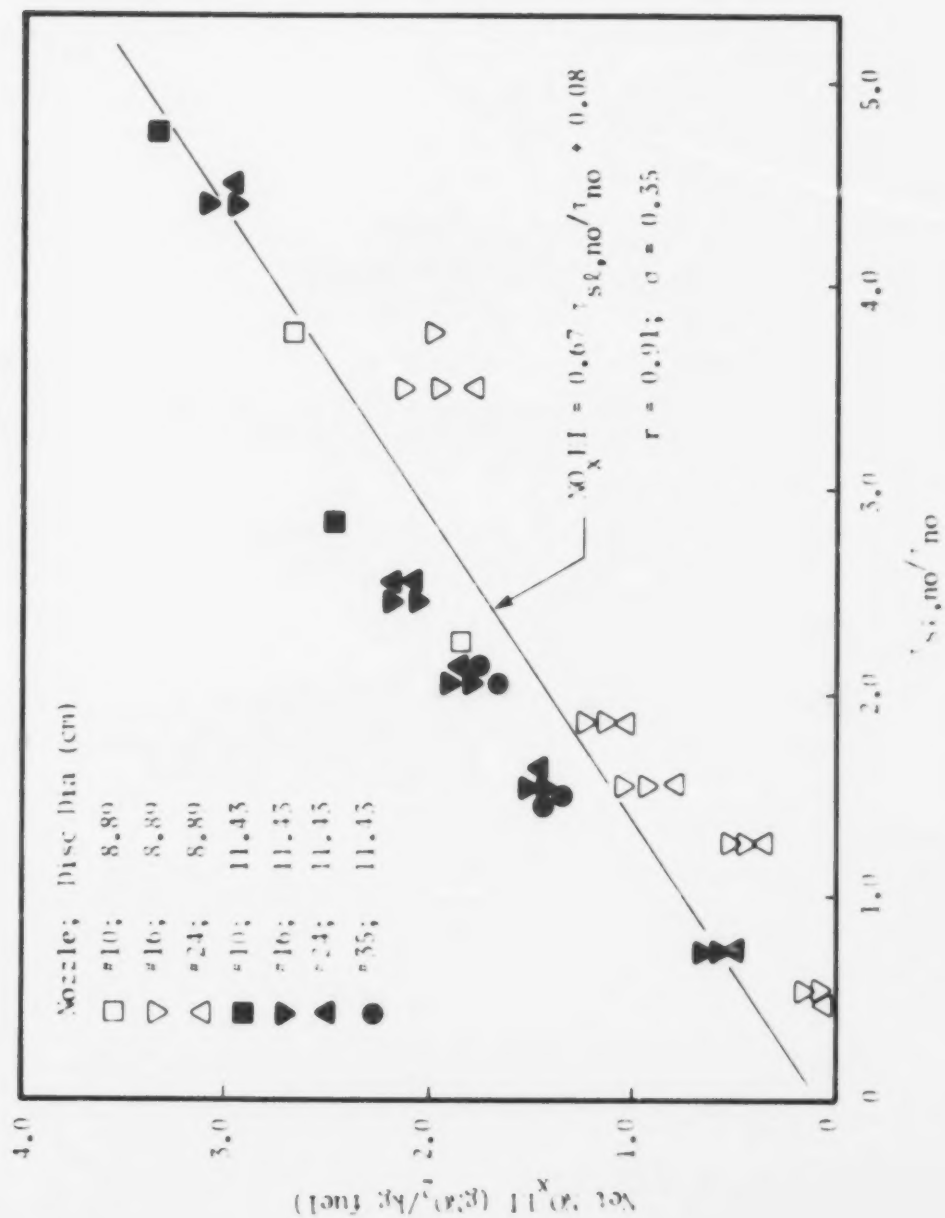
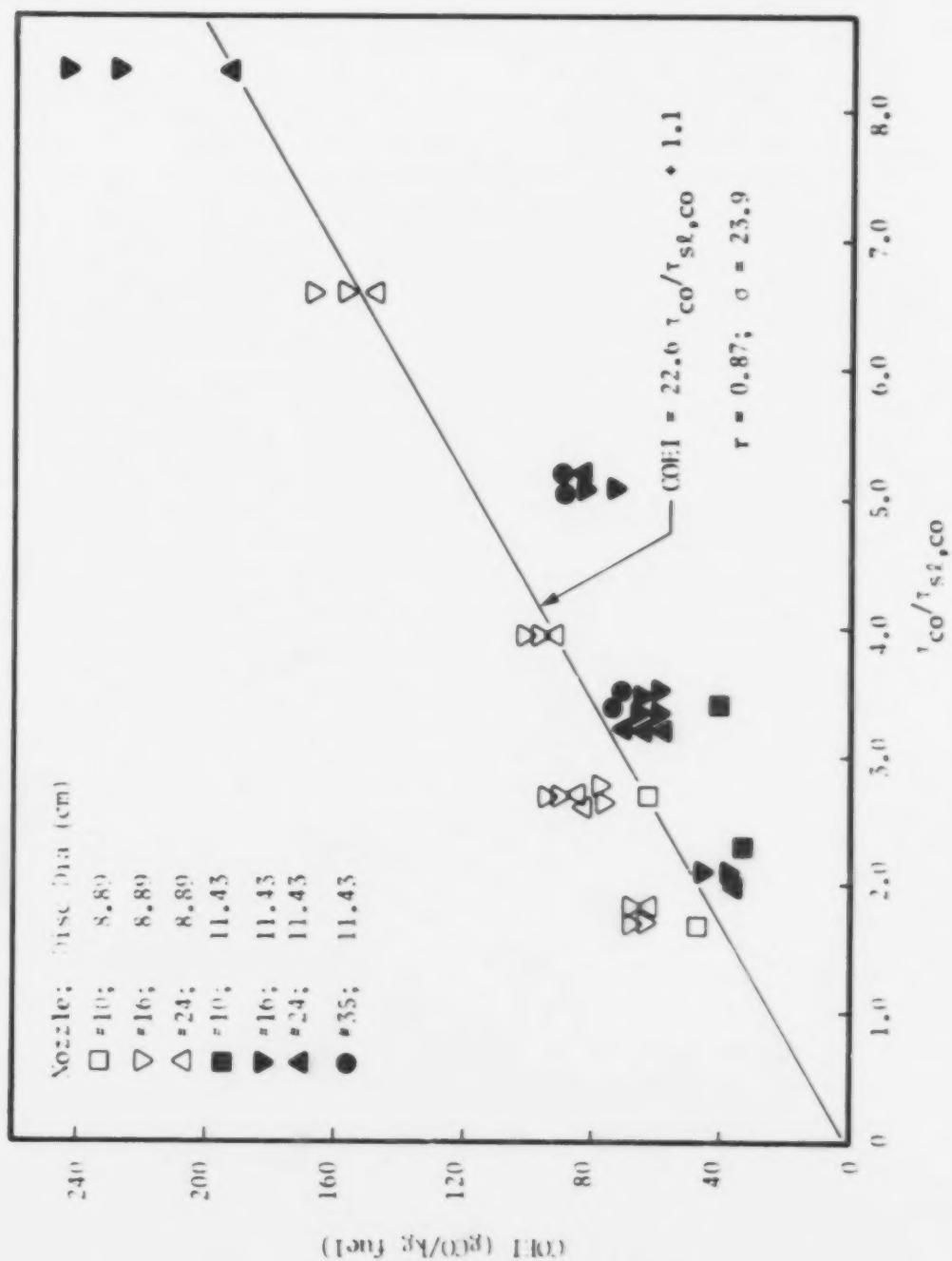


Figure 4-9. W_{x11} versus $t_{s,no}/t_{no}$ (Jet-A fuel)

Figure 4-10. COI versus $T_{CO}/T_{SF,CO}$ (Jet-A fuel)

the CO emissions of both discs were significantly lower. Thus a small percentage of liquid fuel droplets must be penetrating through the shear layer and into the colder free stream at most of the run conditions utilized in the present investigation. Although these droplets are not sufficient in number to substantially affect the NO_x emissions from the larger disc case, they do represent a sizable contribution to both the CO and HC emissions. In particular if these fuel droplets evaporate far downstream in the combustor, the time allowed for complete oxidation of the fuel to CO_2 is considerably reduced over the time allowed when the drops evaporate within the recirculation zone.

The primary effect of increasing the fuel droplet lifetime is then to increase the liquid fuel penetration distance which in turn influences exhaust pollutant emissions. In particular as the droplet lifetime increased, NO_x emissions decreased while CO emissions increased; both effects are attributable to poorer combustion efficiency. Although the fuel penetrating into the free stream may sustain a flame (Fig. 3-4), the gas velocity within the free stream is sufficiently high that rapid cooling of partially burnt gases occurs which results in reduced NO_x but increased CO. The lower NO_x emissions with increasing droplet size also precludes the existence of droplets burning with individual envelope flames since Bracco (1975) showed that in such a case NO_{xII} increases in proportion to the initial drop diameter squared.

However, with the use of Jet-A fuel, some functional dependence of τ_{eb} must be included in the correlating parameters. Since NO_x emissions generally decreased with increasing τ_{eb} , an inverse dependence is suggested. In the denominator of the correlating parameter a sum of τ_{no} and τ_{eb} is taken rather than a product such that the non-dimensionality of the correlating parameter is retained. Similarly, a sum of τ_{co} and τ_{eb} is taken in the CO correlating parameter since a direct dependence between CO emissions and τ_{eb} exists. Thus the two more universal correlating parameters are

$$\text{NO}_{xII} \sim \frac{\tau_{s2,no}}{\tau_{no} + k_1 \tau_{eb}} \quad 4-12$$

$$\text{COEI} \approx \frac{\tau_{\text{co}} + k_2 \tau_{\text{ch}}}{\tau_{\text{st,co}}} \quad 4-15$$

where the constants k_1 and k_2 relate the chemical and heterogeneous processes but have limited physical significance since the values depend upon the assumed values of the pre-exponential factors in the chemical times and also on the initial droplet diameter used in the τ_{ch} calculations. Here the Sauter mean diameter of the spray was used whereas the drops penetrating farthest downstream are those that have an initial diameter much larger than the Sauter mean. Although Equations 4-12 and 4-15 do necessarily include the dependence of droplet penetration upon nozzle size through the calculation of τ_{ch} , these correlating parameters do not include the dependence of the effective droplet penetration distance upon disc diameter. Alternate expressions which do account for this latter effect are included in Appendix B.

The disc combustor NO_x and CO emissions from both the propane and Jet-A experiments are displayed in Figs. 4-11 and 4-12, respectively, versus the appropriate correlating parameter. Also included on Figs. 4-11 and 4-12 are data taken from Dodds et al. (1975) for the same disc burner fueled with Jet-A and at similar pressures and air flows but for inlet air temperatures from 600 to 800°K. The values of the constants k_1 and k_2 indicated on the two respective figures were determined such that the correlation coefficient of each graph was a maximum. The necessary values of the characteristic times were taken from Tables 4-1 through 4-5.

Although some data scatter in Figs. 4-11 and 4-12 is evident, the data generally fall about the suggested linear relationships and the success of the correlating parameters is demonstrated by the fact that the values of the correlation coefficients are both above 0.90. Considering the large number of data points representing wide variations in burner operating points and geometries contained in Figs. 4-11 and 4-12, these high values of the correlation coefficients can be considered to correspond to excellent correlation of the experimental data. In particular the scatter seen in each figure is typical of day to day scatter in measurements gathered at a fixed run condition. Note also

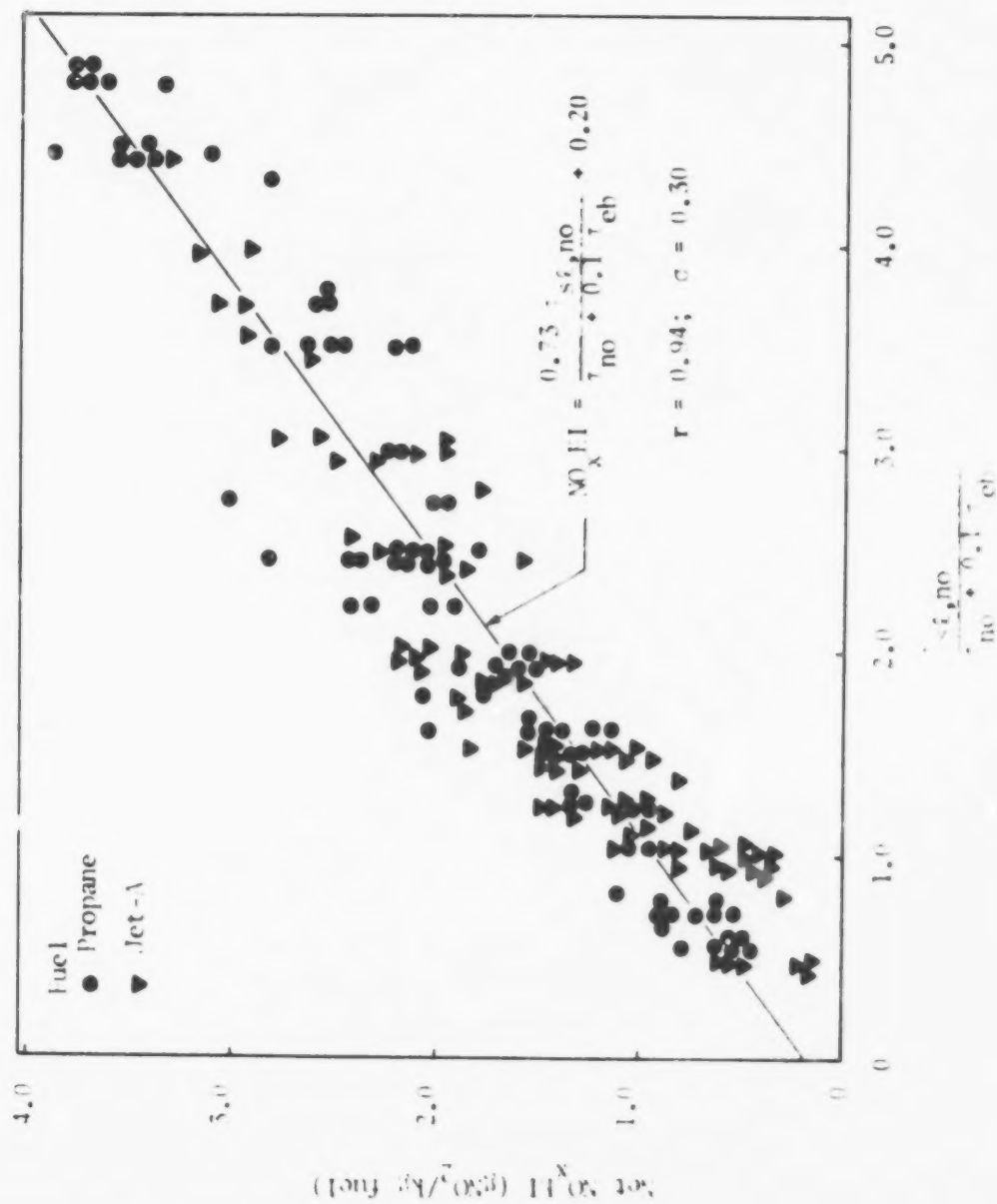


Figure 4-11. Correlation of all disc burner NO_x data

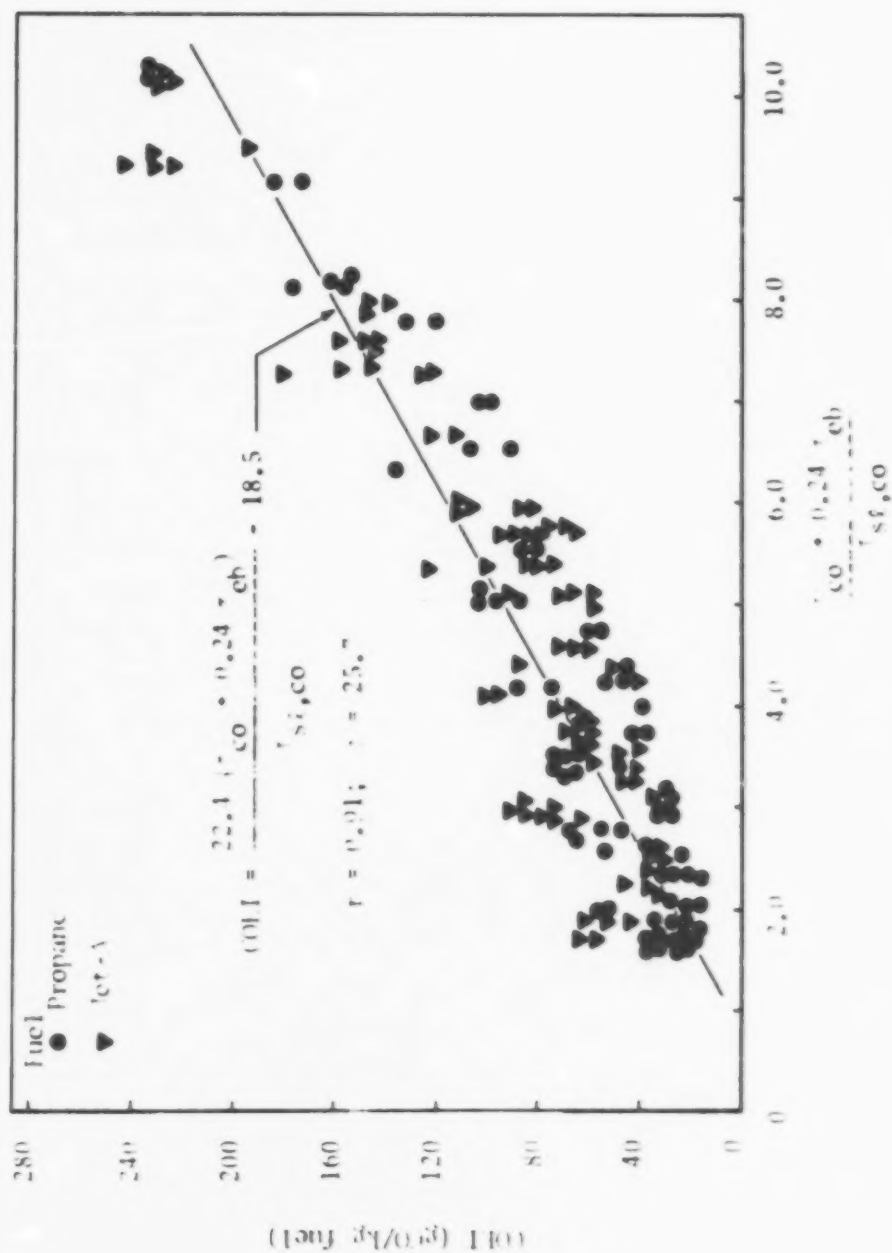


Figure 4-12. Correlation of all disc burner COI data

that the measurements corresponding to propane fuel and Jet-A fuel are overlapped which indicates that the change in fuel type has been successfully incorporated into the correlating parameters (through τ_{no} and τ_{eb}).

The COI data contained in Fig. 4-12 appear to exhibit a somewhat stronger than unity power dependence upon the utilized correlating parameter, suggesting that perhaps a two term approximation to the exponential function (Equation 4-7) is not adequate. However subsequent use of an exponential correlating parameter did not yield an increase in the correlation coefficient displayed on Fig. 4-12. In addition, if one considers only the propane data (Fig. 4-5), a linear dependence of COI upon $\tau_{co}/\tau_{so,co}$ is adequate. Thus the curvature existing in Fig. 4-12 most likely results from the method of including τ_{eb} in the correlating parameter. In addition, the data points to the far right in Fig. 4-12 which fall above the indicated linear relationship are primarily low combustor pressure run points with Jet-A fuel. Perhaps for these points, the effect of low pressure has not been adequately accounted for in the calculation of τ_{eb} and the estimated values are too low.

As was the case with propane fuel, HC and CO emissions from the disc burner fueled with Jet-A correlated well with each other (Fig. 4-10). Run points for which CO emissions were high due to excessive fuel droplet penetration into the free stream also produced high HC emissions. Thus once CO emissions are minimized through analysis of the appropriate correlating parameter, HC emissions will also be minimal.

In the above analysis, all of the obtained emissions data from the disc burner were correlated to a high degree of success by parameters composed of simple but physically reasonable combinations of the characteristic times. The success of the present use of characteristic times in emissions analysis results from the fact that the key physical phenomena leading to burner exhaust emissions have been identified and associated with appropriate time scales. These time scales in turn are composed of combustor operating parameters such as inlet air temperature or air flow rate which have been used in the past to qualitatively describe combustor emissions trends for limited ranges of engine operation.

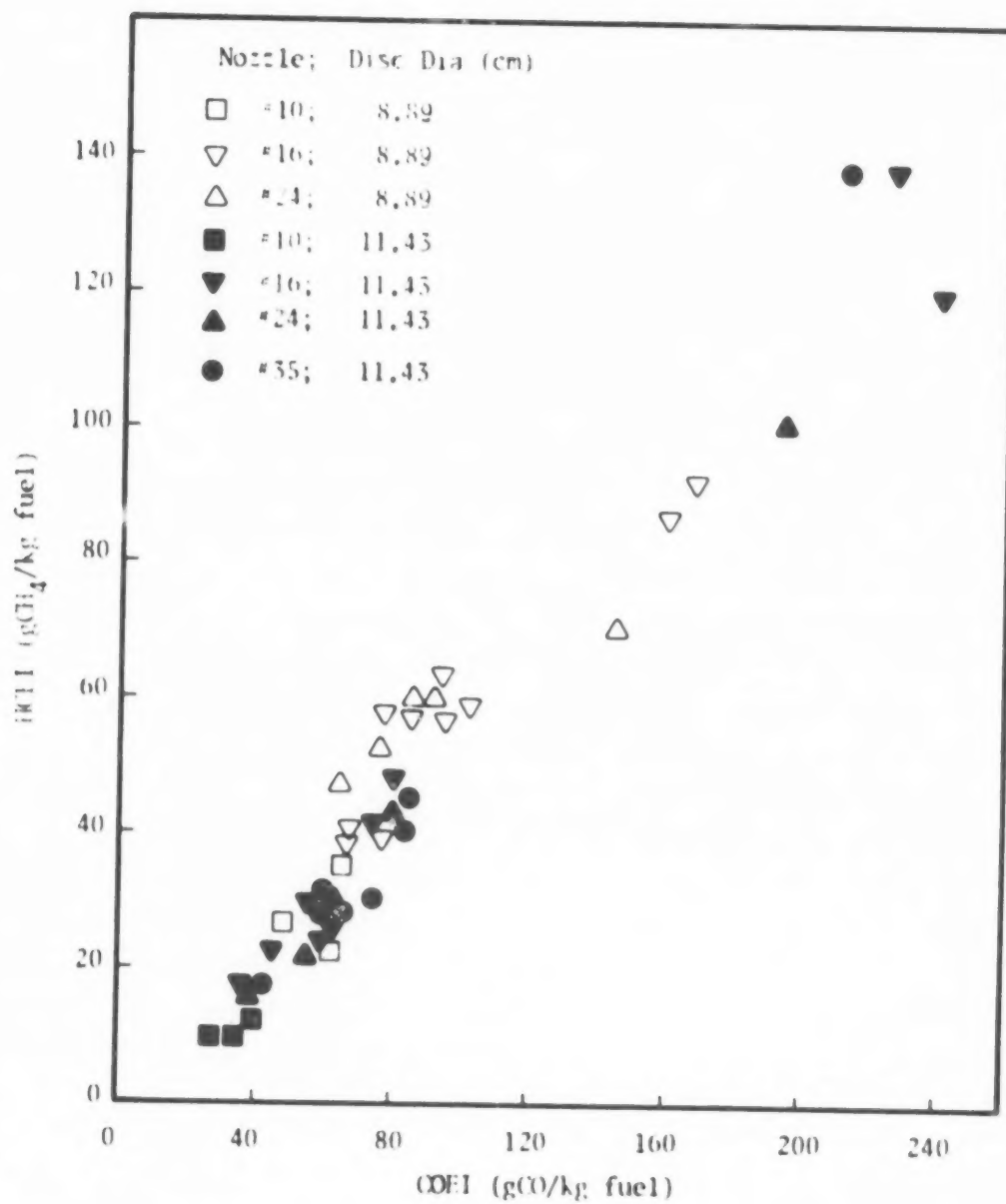


Figure 4-13. HCEI versus COEI (Jet-A fuel)

V. EMISSIONS CORRELATIONS AND SAMPLE OPTIMIZATION: THE GT-309 AUTOMOTIVE GAS TURBINE COMBUSTOR

The imposition of standards (EPA, 1973) and goals (Blazowski and Henderson, 1972) for aircraft pollutant exhaust emissions has stimulated the present investigation into the basic processes occurring in the combustion of a liquid fuel spray. In the previous sections, the time scales developed to characterize the dominant ongoing processes in a gas turbine combustor were quantified for the axisymmetric, turbulent diffusion flame disc burner. In this Section, the emissions correlating parameters developed in Section IV are extended to a conventional automotive gas turbine combustor, the GT-309. Quantification of emissions correlating parameters for this combustor should allow determining if any combustor of the same overall design can meet the emissions standards (Fig. 4-8), not attempted here for the truck/automobile application.

A. GT-309 Cycle and Combustor Designs

The combustor inlet conditions are keyed by percent of design speed for the gasifier turbine (Wade and Cornelius, 1972) and are listed in Table 5-1.

NO emissions in ppm have been reported (Cornelius and Wade, 1970; Wade and Cornelius, 1972; Fredriksen, 1972) for the ten differing geometries of the GT-309 burner listed in Table 5-2. The original nomenclature is preserved, where P2 denotes primary zone. Accompanying Table 5-2 is Fig. 5-1, where the lengths and diameters are defined. The first nine burners are shown schematically in the upper portion of Fig. 5-1; all had a standard dome (except for the rich primary zone burners where some holes were blocked off) and film cooling slot design, a row of primary holes, and a row of secondary holes. The decreased diameter burner (lower part of Fig. 5-1) had a different dome design and only one row of holes. t_{no} and t_{co} in Table 5-2 are the macroscales for the respective mixing times, to be defined later.

Table 5-1

(3)-500 Combustor Operating Conditions
(from Kade and Cornelius, 1972)

Compressor Inlet Speed	Inlet Air Temperature, $^{\circ}\text{K}$	Inlet Pres- sure, atm	Air Flow Rate, kg/sec	Fuel Flow Rate, kg/hr	Computed T_c , $^{\circ}\text{K}$	τ_{no} , msec	T_b , $^{\circ}\text{K}$	τ_{co} , msec
100	862	3.08	1.66	59.8	2555	.295	1200	2.55
90	807	3.08	1.59	45.0	2542	.275	1200	2.55
80	741	2.55	1.12	51.5	2555	.259	1200	2.55
70	686	2.08	0.99	21.0	2558	.231	1200	2.55
60	757	1.70	0.81	10.5	2465	.655	884	8.01
50	701	1.44	0.62	6.1	2469	.504	889	7.82

Table 5-2

(1-509 Dimensions (cm and cm²) (see Figure 1)
(from Cornelius, 1976)

Barrier Designation	L _{dome}	d _{dome} ^a	L _{film}	d _{film}	L _{pri}	d _{pri} ^a	L _{sec}	d _{sec} ^a	A _{holes}	L _{comb}	d _{comb}	L _{no}	L _{co} ^b
Standard	1.55	0.60(36)	5.40	0.58	8.69	1.00(6)	20.57	5.18(8)	108.58	26.54	19.05	9.89	9.89 5.97
Early Quench #1	1.55	0.60(36)	5.40	0.58	8.69	1.00(6)	15.59	5.18(8)	108.58	26.54	19.05	7.95	----
Early Quench #2	1.55	0.60(36)	5.40	0.58	4.52	1.00(6)	9.22	5.18(8)	108.58	26.54	19.05	6.21	6.21 2.89
Lean PZ	1.55	0.60(36)	5.40	0.58	8.69	1.00(6)	44.07	5.18(8)	108.58	45.59	19.05	15.50	----
Lean PZ	1.55	0.60(36)	5.40	0.58	8.69	1.77(12)	20.57	5.51(4)	108.52	26.54	19.05	6.21	9.89 2.89
Rich PZ	0.78	0.60(12)	5.40	0.58	8.69	1.00(6)	20.57	5.55(8)	108.15	26.54	19.05	9.89	9.89 2.89
Lean PZ & Early #1	1.55	0.60(36)	5.40	0.58	8.69	1.77(12)	15.59	5.51(4)	108.52	26.54	19.05	6.21	----
Rich PZ & Early #1	0.78	0.60(12)	5.40	0.58	8.69	1.00(6)	15.59	5.55(8)	108.15	26.54	19.05	7.95	----
Lean PZ & Early #2	1.55	0.60(36)	5.40	0.58	4.52	1.77(12)	9.22	5.51(4)	108.52	26.54	19.05	6.21	----
Decreased Diameter	0	0.55(18)	2.79	0.64	----	-----	7.14	1.93(12)	62.75	18.57	15.97	4.75	4.75 2.55

^aNumber in parentheses is number of holes of stated diameter.

^bFirst number is high power (generally 70-100%) L_{co}; second number is low power (generally 50-60%) L_{co}. A blank indicates no CO emissions data were available.

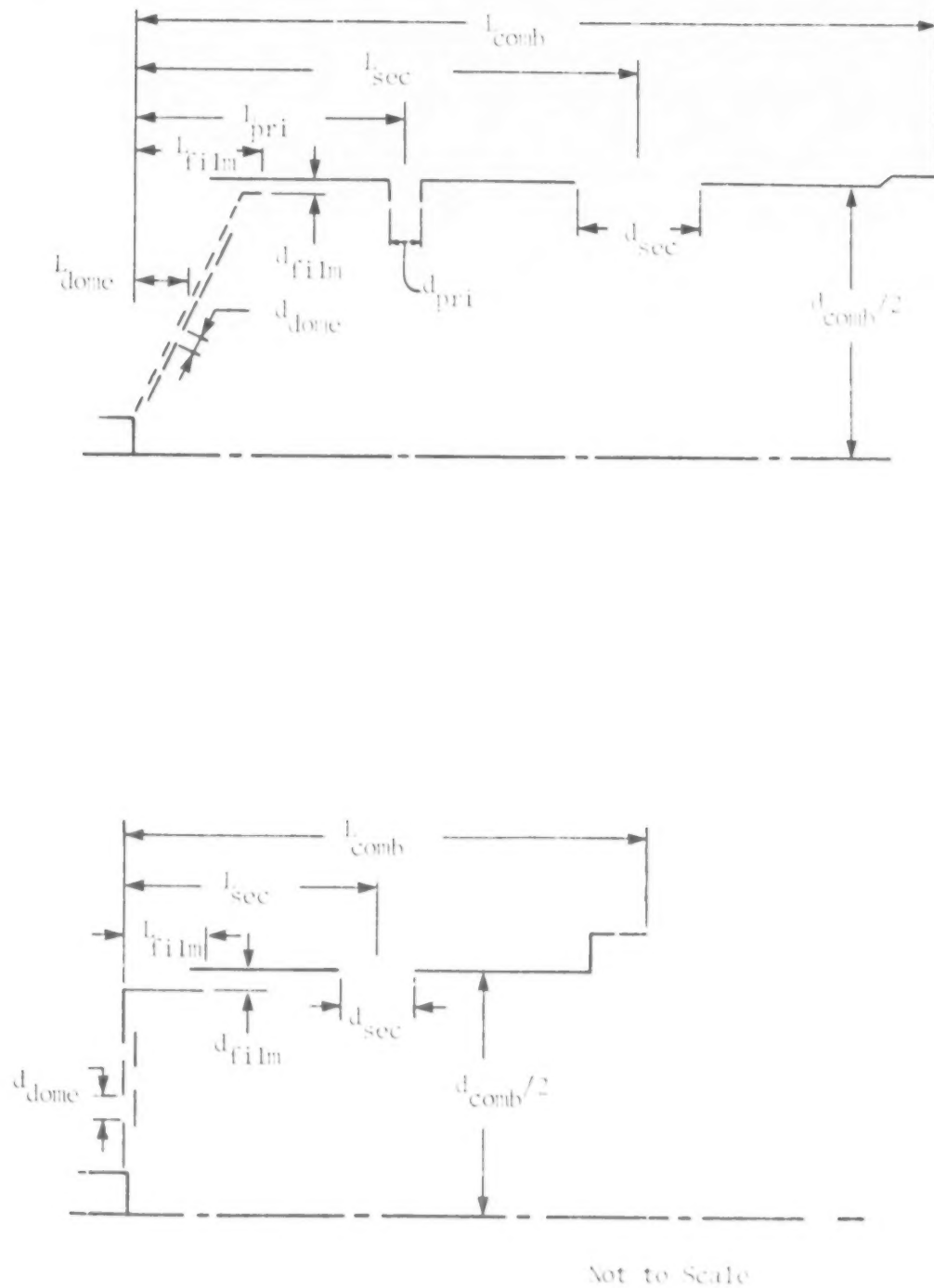


Figure 5-1. Schematic diagrams of the various GT-509 burners. Dimensions for particular configurations (Cornelius, 1976) are listed in Table 5-2.

CO and HC emissions were reported for the standard burner only (Wade and Cornelius, 1972). Through the kind assistance of General Motors personnel the data presented in Table 5-3 were provided (Cornelius, 1976).

Table 5-3
GT-309 CO and HC Exhaust Emissions
(from Cornelius, 1976)

Burner Designation	% of Design Gasifier Speed	CO, ppm	HC as C ₆ H ₁₄ , ppm
Lean PZ	70-100	12(ave) ^a	0.2(ave) ^a
	60	90	1.7
Rich PZ	70-100	33(ave) ^a	0.1(ave) ^a
	60	91	0.75
Early #2	70-100	35(ave) ^a	0.3(ave) ^a
	60	16	2.9
Decreased Diameter	100	69	~ 0
	90	77	~ 0
	80	87	~ 0
	70	75	~ 0
	60	95	~ 0
	50	150	1.4

^aIn subsequent conversion to emissions indices the 90% power air and fuel flow rates (Table 5-1) were utilized.

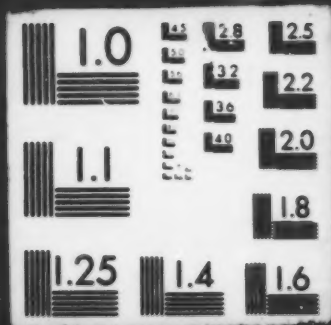
B. Emissions Correlating Parameters

In the correlation of the pollutant emissions of the air-fueled disc burner, droplet vaporization effects were accounted for by introducing ϕ_d into the correlation parameters. Analytical studies on negligible droplet effects (Equations 3-12 and 3-13, Section 3.3)

2 0 F 2

P B

258 269



dealing with gas turbine spray flames of conventional fuels (here diesel #2) it is logical as a first step to ignore heterogeneous processes: this was done here, and some justification is provided by the NO_x EI correlation below, as well as the apparent mixing control at the idle condition where droplet evaporation is expected to be most important (Mellor, 1973).

Mixing times (τ_{sl}) in the shear layer separating downstream flow from the flameholding recirculating flow in the combustor primary zone should be evaluated as the ratio of local turbulence integral length scale to turbulent intensity. Since this information is rarely available for combusting flows, the τ_{sl} are again taken as proportional to a significant macrodimension and the reciprocal of a time-mean velocity. As was the case for the disc burner, CO and NO emissions from the GT-509 combustor are expected to originate in separate regions of the shear layer, so that $\tau_{sl,co}$ and $\tau_{sl,no}$ are not identical.

Chemical kinetic times for CO oxidation (τ_{co}) and NO formation (τ_{no}) are based on the variation of burner exhaust emissions with inlet air temperature for activation energy (Section II.C); pre-exponential factors are chosen to make τ_{co} and τ_{no} of the same order as $\tau_{sl,co}$ and $\tau_{sl,no}$. For the results reported in this section

$$\tau_{no} = 10^{-12} e^{153000 \text{ cal/mole}/RT_f} \text{ msec} \quad 5-1$$

$$\tau_{co} = 10^{-1} e^{7700 \text{ cal/mole}/RT_b} \text{ msec} \quad 5-2$$

R is the universal gas constant, T_f is the adiabatic stoichiometric flame temperature based on air inlet temperature and pressure, and T_b is the adiabatic burned gas temperature based on overall combustor equivalence ratio. Note that once the engine cycle operating points and the fuel are specified, τ_{co} and τ_{no} are known parameters.

As noted above, fuel droplet effects are assumed to be negligible in a well designed gas turbine combustor. Accordingly, the following correlations are sought:

$$\text{NO}_x \text{ EI} \approx \tau_{sl,no} / \tau_{no} \quad 5-3$$

$$\text{COEI} \sim \tau_{\text{co}} / \tau_{\text{sl,co}}$$

5-4

Since τ_{no} and τ_{co} are known, the task is to determine how the two mixing times depend on the burner geometry.

C. Correlation of Exhaust Emissions

1. Effect of Power Setting on Nitric Oxide

Typical NO levels in ppm for five different GT-309 combustors are reproduced from Wade and Cornelius (1972) in Fig. 5-2. As engine power is increased from idle (50 and 60%) to design, the nitric oxide mole fraction increases; the NO_xEI exhibits a more complicated behavior since it includes both the air and fuel flow rates: the peak emission of NO_x in g NO_x/kg fuel is found at 70% power.

2. Effect of Air Inlet Temperature on Nitric Oxide

Lipfert (1972) has shown that NO_xEI correlates on a universal curve versus inlet air temperature for conventional combustors, and Sawyer et al. (1975) redrew the Lipfert plot (Fig. 2-2), obtaining an activation energy of 156,000 cal/mole. A plot of NO_xEI versus $1/\tau_{\text{no}}$ is essentially equivalent, and is shown in Fig. 5-3 for the same five GT-309 burners. τ_{no} correlates the variation in NO_xEI with air inlet temperature (or percent power) well for a particular combustor, but the slope of the correlation is a function of burner geometry, inconvenient for the combustor designer.

3. Effect of Burner Geometry on Nitric Oxide

To include burner geometry in the NO_xEI correlation requires the determination of $\tau_{\text{sl,no}}$, the time near-stoichiometric eddies in the combustor are allowed to form NO before quenching. Concentration measurements within various gas turbine burners have shown that these NO-forming eddies are present in the downstream end of the shear layer surrounding the centerline recirculation zone (Mellor, 1976; Tuttle et al., 1976). The GT-309 combustor is characterized by large secondary air penetration jets (Fig. 5-1), which most likely terminate the recirculation zone as they force the shear layer to the centerline; they also therefore quench

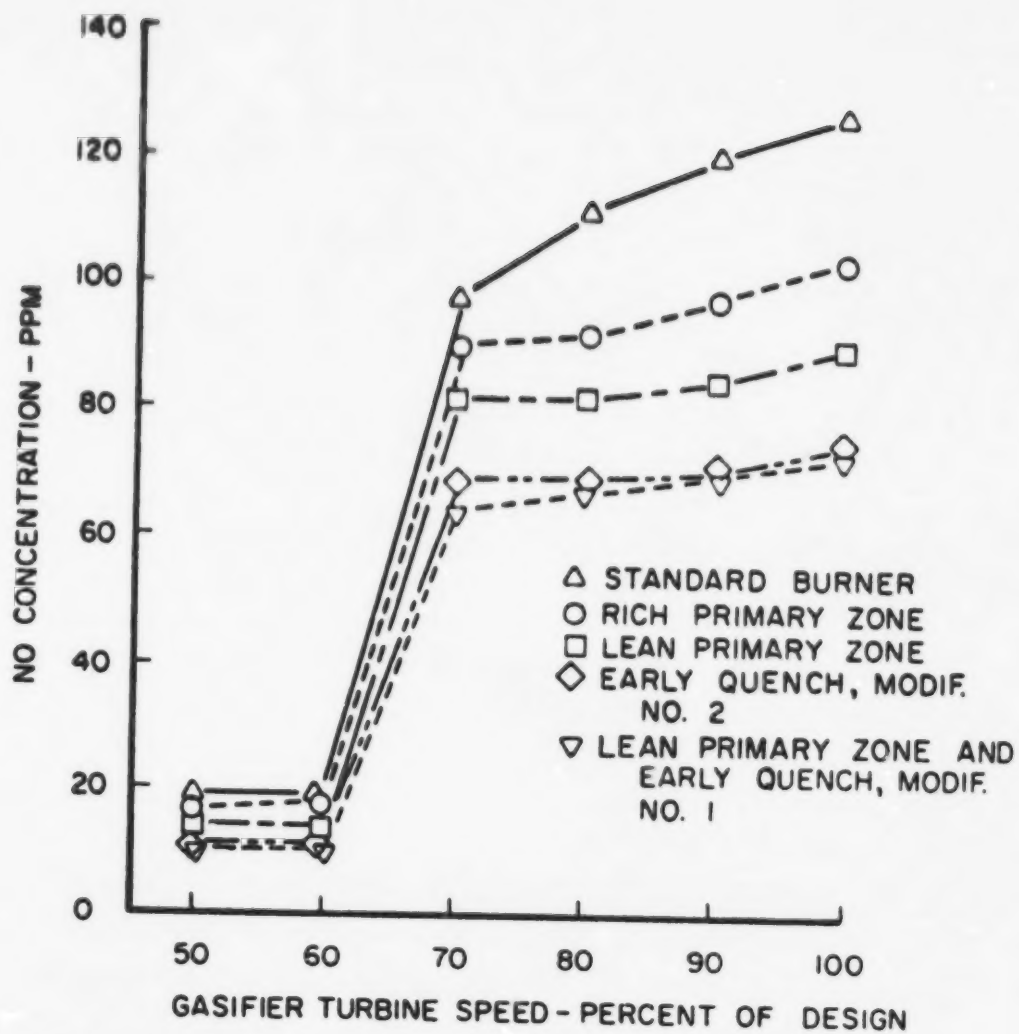


Figure 5-2. NO (ppm) versus percent of design power for five GT-309 burners (from Wade and Cornelius, 1972)

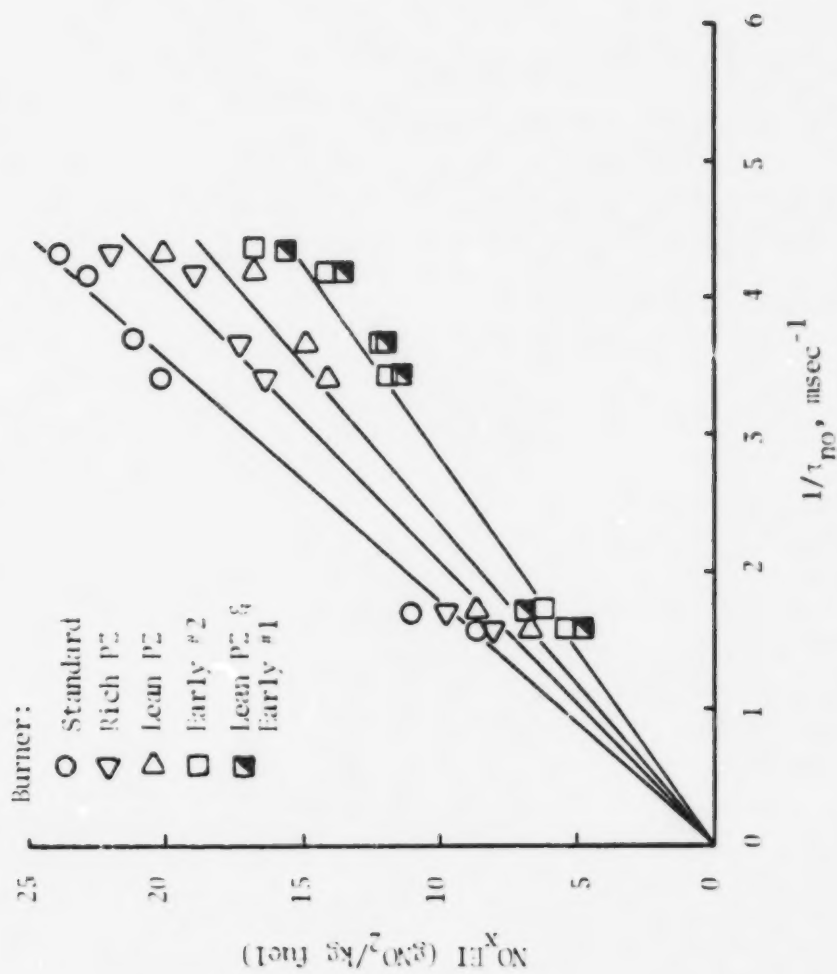


Figure 5-3. NO_x versus $1/\tau_{no}$ for five GT-309 burners

NO formation. Consequently the macroscale for NO formation L_{no} is defined as

$$L_{no}^{-1} = L_{sec}^{-1} + d_{comb}^{-1} \quad 5-5$$

where inclusion of d_{comb} delineates the radial extent of the shear layer. Computed L_{no} are listed in Table 5-2 for the various burners. Then

$$\tau_{sl,no} = L_{no}/V_{\phi=1} \quad 5-6$$

Here $V_{\phi=1}$ is the flow velocity in the stoichiometric region of the combustor flow, assumed proportional to the turbulent intensity in that region.

This velocity is taken as

$$V_{\phi=1} = \dot{m}_{a\phi=1} RT_f / pM \frac{\pi}{4} d_{comb}^2 \quad 5-7$$

p is the combustor inlet pressure, M is the flow molecular weight, taken as that of pure air for convenience, and $\dot{m}_{a\phi=1}$ is the primary zone air flow rate corrected to stoichiometric at design power. For a particular engine power it is calculated as

$$\dot{m}_{a\phi=1} = .8 \times .185 \dot{m}_a \quad 5-8$$

where $.185 \dot{m}_a$ is the primary zone airflow rate and $.8$ is the estimated primary zone equivalence ratio for the standard burner at 100% power (Wade and Cornelius, 1972).^{*} Note that $\dot{m}_{a\phi=1}$ assumed independent of combustor geometry provides the best NO_xEI correlation (see below). Except for the cross-sectional area effect $V_{\phi=1}$ is then completely specified by the combustor inlet conditions.

NO_xEI is displayed versus $\tau_{sl,no}/\tau_{no}$ in Fig. 5-4 for ten different burners and three different fuel injectors (see Cornelius and

^{*} $\dot{m}_{a\phi=1}$ could equally well be defined as the primary zone airflow rate. The slope of the subsequent NO_xEI correlation would simply increase by $1/.8 = 1.25$.

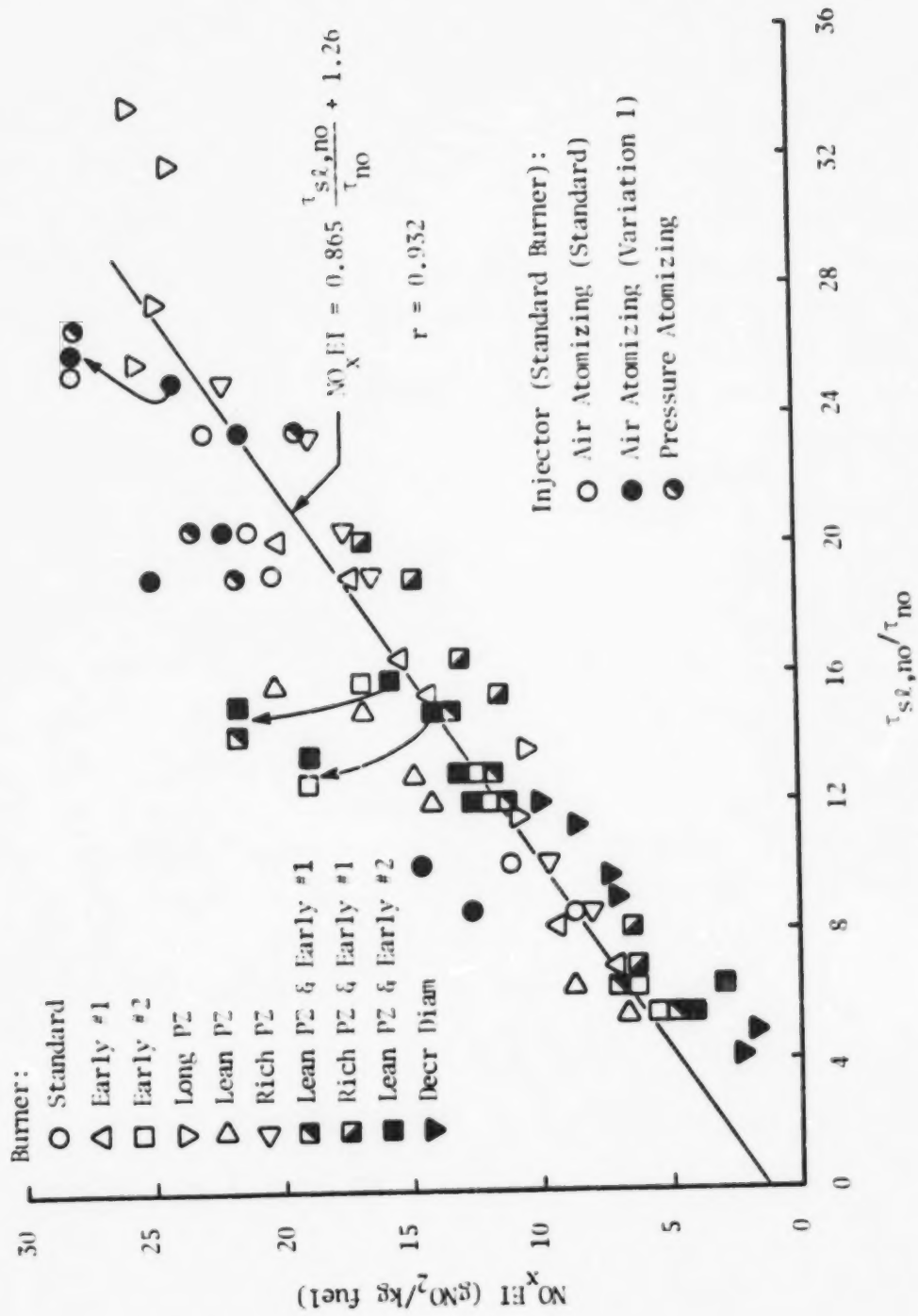


Figure 5-4. $\text{NO}_x \text{EI}$ versus $\tau_{sl,no}/\tau_{no}$ for ten GT-309 burners and three fuel injectors

Wade (1970) for descriptions of same). The inclusion of the geometry effect is shown by a comparison of Fig. 5-3 and 5-4. A least squares fit of the correlation provides the straight line shown in Fig. 5-4 and the equation

$$\text{NO}_x \text{EI} = 0.865 \tau_{st, \text{no}} / \tau_{\text{no}} + 1.26 \quad 5-9$$

with a correlation coefficient of 0.932. The lack of any dependence upon fuel injector (within the data scatter) confirms that for all injectors utilized (here with diesel fuel #2) $\tau_{\text{eb}} \ll \tau_{st, \text{no}}$ (i.e., the spray flame is mixing, not evaporation controlled; see also Mellor, 1973).

Changes in combustor geometry thus affect $\text{NO}_x \text{EI}$ primarily through L_{no} , with decreasing L_{no} (decreasing L_{sec} or d_{comb}) decreasing $\text{NO}_x \text{EI}$. The open inverted triangles to the right of the correlation in Fig. 5-4 are for a combustor with an elongated primary zone and can be moved to the curve by the reasonable assumption that at intermediate power points the actual flame length is less than L_{sec} .

4. Effect of Burner Geometry on Carbon Monoxide

CO emissions originate in the outer regions of a spray flame where mixing with colder inlet air terminates CO oxidation (Mellor, 1976). In a combustor the discrete addition of air through holes and slots provides this quenching. The turbulent intensity is taken proportional to the mean air injection velocity through the various openings in the dome and liner.

$$V_{a, \text{hole}} = \dot{m}_a RT_i / p A_{\text{holes}} \quad 5-10$$

Here all symbols have their previous meanings, T_i is the inlet air temperature, and A_{holes} is the total area of holes and slots (see Table 5-2).

To correlate the measured CO emissions it was found necessary to allow L_{CO} , the CO macroscale, to decrease with decreasing engine power (see Table 5-2). In particular, idle CO emissions are quenched and thus

originate in more upstream air addition positions than those nearer design operation. L_{CO} is defined as

$$L_{CO}^{-1} = L_{quench}^{-1} + d_{comb}^{-1} \quad 5-11$$

where for 70 to 100% power $L_{quench} = L_{sec}$ and for 50 and 60% $L_{quench} = L_{dome}$, L_{film} , or L_{pri} , depending on combustor design.

The correlation is shown in Fig. 5-5 for the five combustors for which data are available.

$$COEI = 0.947 \tau_{CO} / \tau_{sl,CO} - 0.64 \quad 5-12$$

where

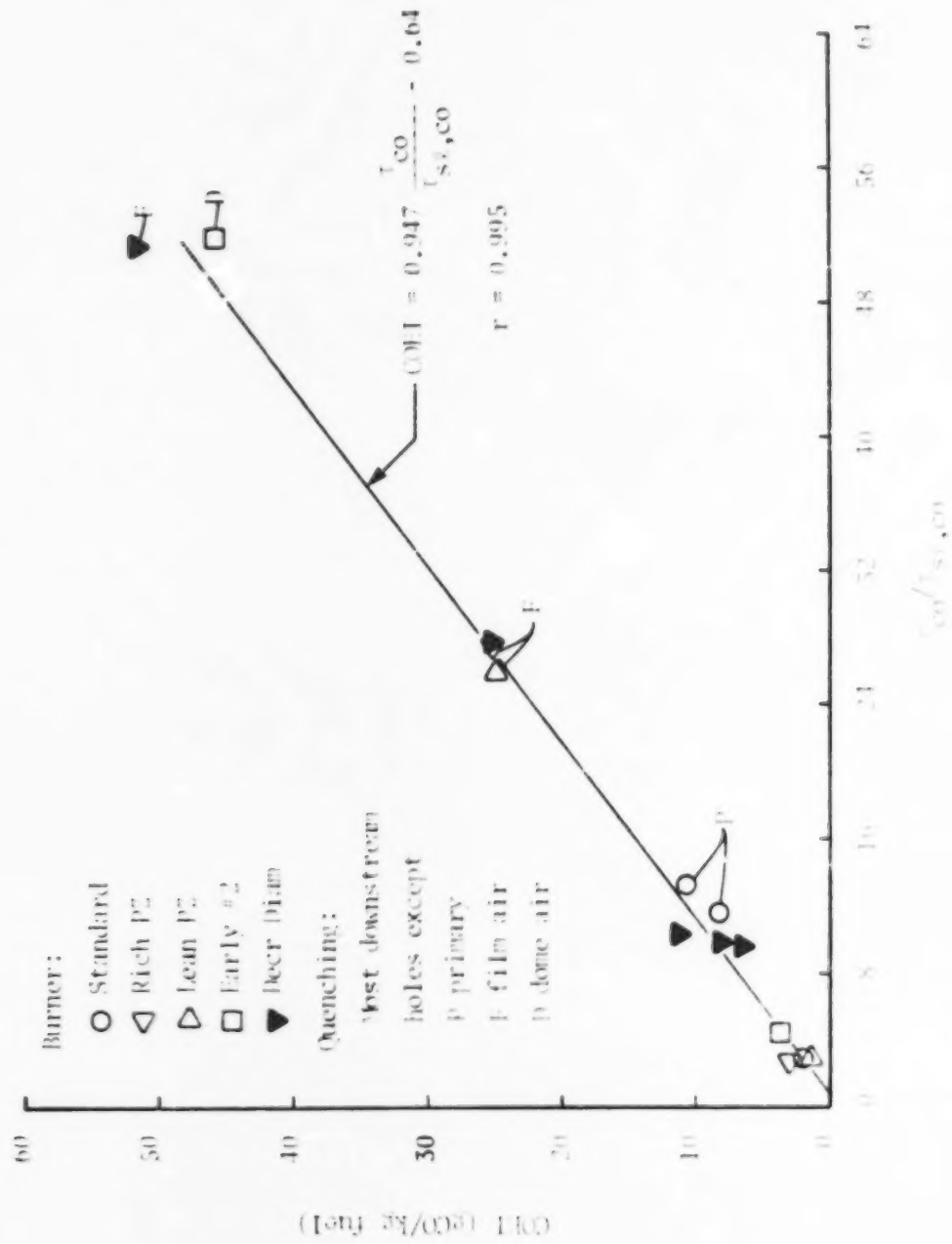
$$\tau_{sl,CO} \equiv L_{CO} / V_{a,hole} \quad 5-13$$

with a correlation coefficient of 0.995. Use of the exponential correlating parameter for CO emissions (Equation 4-7, $k = 34$) in place of the linear parameter $\tau_{CO} / \tau_{sl,CO}$ yielded no increase in the correlation coefficient. If L_{CO} was not allowed to decrease with decreasing engine power, the correlation coefficient for an exponential correlation of the data was reduced to 0.777.

As noted in Fig. 5-5, high power CO quenching always occurs in the most downstream air penetration jets. At idle, CO can originate in the region of the dome air holes (early quench #2), at the film cooling slot (decreased diameter, rich PZ, and lean PZ), or in the primary air addition holes (standard combustor). Note that the finding that L_{CO} increases with increasing engine power is entirely consistent with the expected variation in flame length with engine power, and that a correlation independent of droplet lifetime τ_{eb} again reaffirms the unimportance of heterogeneous effects.

5. Unburned Hydrocarbons and Carbon Monoxide

If HC/EI can be shown proportional to COEI, then the above COEI equation can be used to estimate unburned hydrocarbon emissions. A graph

Figure 2-3. COI versus CO/CO₂ for five 61-309 burners

of HCEI versus COEI for the GT-309 burners demonstrates that the proportionality constant is a function of combustor geometry. HCEI versus COEI ($L_{co}/L_{co, std}$), where $L_{co, std}$ is the CO macroscale for the standard burner as a function of engine power shown in Table 5-2, improves the universality of the correlation, but too few data are available (Table 5-3) to provide a convincing test. Generally however HCEI is proportional to COEI (Lipfert, 1972; Tuttle et al., 1975).

6. Summary

With statistically high levels of confidence, the following correlations have been developed for emissions from various GT-309 burners

$$NO_{x,EI} = \{0.865 (L_{sec}^{-1} + d_{comb}^{-1})^{-1} pM \frac{\pi}{4} d_{comb}^2 \} : \quad 5-14$$

$$\{ .8 \times .185 \dot{m}_a RT_f 10^{-15} e^{133000 \text{ cal/mole}/RT_f} + 1.26$$

$$COEI = \{0.947 \times 10^{-4} e^{7700 \text{ cal/mole}/RT_b} \dot{m}_a RT_i \} :$$

$$\{ pM_{holes} (L_{quench}^{-1} + d_{comb}^{-1})^{-1} \} - 0.64 \quad 5-15$$

Based on a characteristic time model of the spray combustion process, these equations include the effects of changes in both burner geometry and inlet conditions. They can also be used to predict emissions from geometrically similar burners (provided that fuel droplet vaporization remains negligibly fast).

D. Optimum Emissions Predicted for Geometrically Similar Combustors

In Fig. 5-6 predicted $NO_{x,EI}$ and COEI are shown as solid curves versus L_{sec}/d_{comb} , the distance from the dome to the secondary air addition holes non-dimensionalized with burner diameter. The figure was computed for the 90% power point and for diameters of the burners actually tested at General Motors; the test results (which were used to develop the above correlations) for the various combustors are shown as discrete symbols connected by a line to the corresponding prediction.

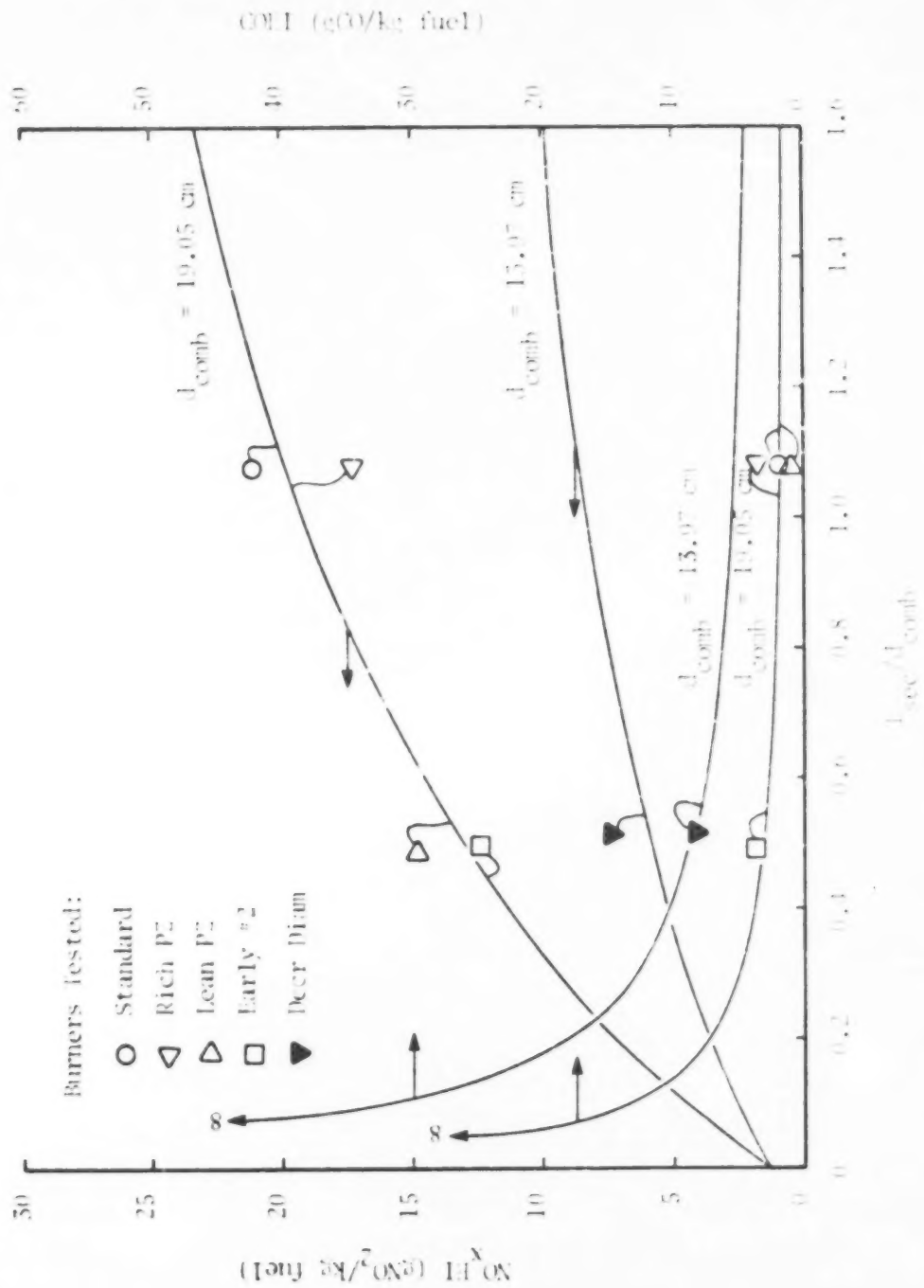


Figure 5-6. Predicted $\text{NO}_x \text{ FI}$ and COI versus $U_{\text{sec}}/d_{\text{comb}}$ for combustors of the GL-300 engine at 90% of design power.

Figure 5-6 shows that burner diameter has a stronger effect on NO_xEI than COEI at near design power levels, suggesting that some decrease in NO_xEI could be obtained without too large an increase in high power COEI by choosing a smaller burner diameter. Similarly $L_{\text{sec}}/d_{\text{comb}}$ can be reduced to about 0.3 before a precipitous increase in COEI occurs. The decrease in NO_xEI (and increase in COEI) with decreasing $L_{\text{sec}}/d_{\text{comb}}$ reflects the benefit (disadvantage) of early quenching in halting NO formation (CO oxidation) at the high engine powers. The diameter effect at constant $L_{\text{sec}}/d_{\text{comb}}$ results from the same process. Results of this type are well known to combustor designers: the advantage of the characteristic time correlation is that the magnitude of emissions changes from geometry (or combustor inlet condition) alterations are given qualitatively.

A similar prediction of 60% power, idle emissions is displayed in Fig. 5-7. The geometry parameter here is $L_{\text{quench}}/d_{\text{comb}}$, where as noted in a previous section and in the figure, L_{quench} depends upon combustor design. Since NO quenching always occurs at the secondary holes, whereas low power CO emissions result from upstream quenching, NO_xEI are constant, independent of $L_{\text{quench}}/d_{\text{comb}}$. Thus the prediction is that low power combustion inefficiency can be decreased by appropriate geometry changes (increased L_{quench}) with no change in low or high power NO emissions. If high power NO_xEI are decreased by decreasing d_{comb} , then Fig. 5-7 shows the penalty expected for idle COEI.

In the next sub-section these curves are used to design a low NO_x burner of the GT-309 class.

E. Design of a Sample Optimum Burner

To maintain the success of the GT-309 dome and film cooling slot design in eliminating any heat transfer difficulties in the front end of the burner, their design will not be considered variable in the following. For this sample design the combustor diameter will also be held constant at 19.05 cm identical to the standard burner. However, the row of primary holes will be eliminated so that the secondary holes can be moved upstream to minimize NO_xEI at high power. This will require eight 3.35 cm dilution holes to accommodate the air flow rate and maintain

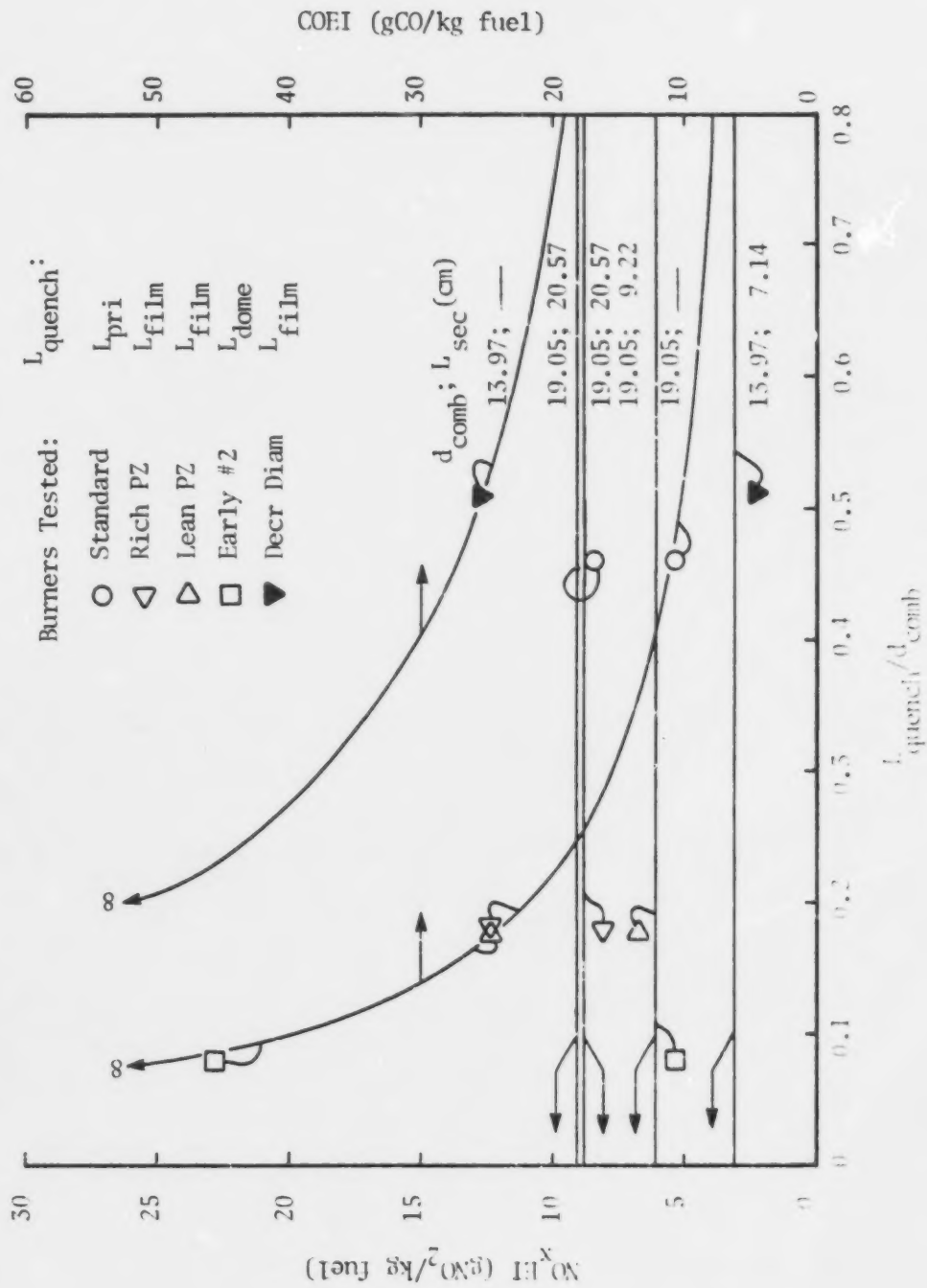


Figure 5-7. Predicted NO_x and COEI versus L_{quench}/d_{comb} for combustors of the CF-509 engine at 60% of design power.

the same air velocity through the holes (recall that increasing this velocity increases COEI).

In the standard combustor the film cooling slot is at $L_{\text{film}} = 5.4$ cm. Allowing 1.0 cm between the slot exit on the dome and the upstream edge of the 1.68 cm radius dilution holes, their centerlines are placed at $L_{\text{sec}} = 6.09$ cm or $L_{\text{sec}}/d_{\text{comb}} = 0.52$. L_{no} is then computed as 4.61 cm, the same as L_{co} at high power. Depending on whether low power CO quenching will occur at the film cooling slot or at the dilution holes, L_{co} at low power will be either 2.89 or 4.61 cm, respectively. From the correlations listed previously, NO_xEI is predicted to be 47% of that from the standard burner over the entire engine operating range, with a corresponding increase in high power COEI of 215%. The increase at idle depends upon where the quenching occurs; if in the film cooling slots a 207% increase over the standard burner could occur.

Exhaust emissions in ppm versus engine power are shown in Fig. 5-8 for the standard burner, the optimal low NO_x burner designed above, and the decreased diameter burner (which is geometrically similar to the optimum). Two CO curves are shown for the optimum burner at idle for quenching in the film cooling slot (F) or in the dilution air (P): if the latter occurs, then the increase in CO at idle is only about 1.5 over the standard burner. The NO emissions for the optimum are almost as low as those from the decreased diameter burner, without the very large CO emissions found for that design. The CO emissions from the optimum could be reduced further by increasing the total hole area for air injection (twelve rather than eight 3.35 cm dilution holes, for example, should reduce the COEI by 32%).

A systematic study of all of the tradeoffs implied by the NO_xEI and COEI correlations would rapidly enable the designer to find the true optimum. The above analysis is offered merely as a sample to obtain a low NO_x burner and show how the characteristic time model can be used.

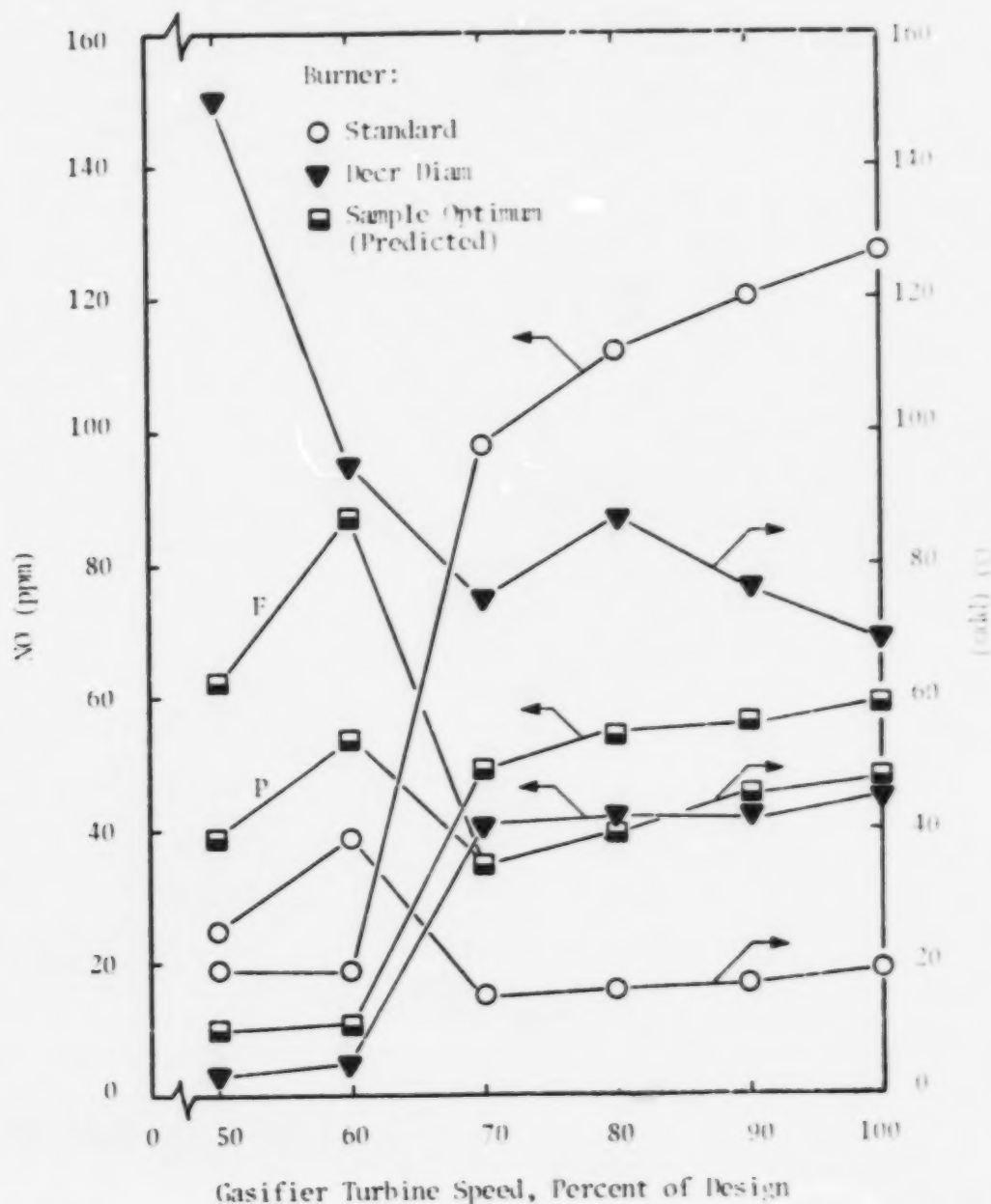


Figure 5-8. Comparison of NO and CO emissions for the standard, decreased diameter, and a sample optimum burner. The curves at idle for the sample represent CO quenching at the film cooling slot (F) or the dilution holes (P).

VI. CONCLUSIONS

The complex physical processes occurring both within a conventional gas turbine combustor and within the disc stabilized flame burner have been considered in terms of the basic subprocesses which constitute the overall combustion phenomenon. Experimental analysis of the spray combustion process via variation of the introduced characteristic times was shown to lead directly to pollutant exhaust emissions correlating parameters. Specifically, a physical model of the spray combustion process was used to guide and motivate the combination of the characteristic times to form non-dimensional correlating parameters. The success of these parameters was demonstrated by the correlation of both CO and NO_x emissions from the disc stabilized flame for various burner geometries, a wide range of burner operating conditions, fuel spray atomization, and two non-similar fuels.

The CO and NO_x emissions from the disc combustor were found to be primarily mixing controlled due to quenching of the CO oxidation and NO formation reactions through dilution of hot reacting gases by colder species. The experiments upon the toothed disc flame stabilizers further substantiates the result of Tuttle et al. (1976) that CO and NO_x emissions are related to the mixing processes in separate regions of the flame structure. CO oxidation reactions were found to be most efficiently quenched in the outer shear layer near the rim of the disc whereas NO formation was found to relate to the mixing processes in the downstream portion of the shear layer near the burner centerline.

The fuel droplet lifetime was used successfully as a criterion for determining the importance of heterogeneous effects upon the flame structure. If the calculated value of this time was less than one millisecond, then liquid fuel droplet effects were observed to be negligible. For longer values of the fuel droplet lifetime, the fuel spray penetrates further into the flow field which increases the shear layer flame length

and/or may establish a free stream flame zone which is stabilized by the spray sheath of the droplets penetrating into the free stream. In this latter case, the combustion efficiency of the burner deteriorated and it was necessary to include a functional dependence of the fuel droplet lifetime in the developed parameters for correlating burner emissions.

For stable burner operation, existing calculations of the time required for the initial breakdown of the hydrocarbon fuel to form CO showed this chemical process to be fast compared to the mixing processes and the subsequent oxidation of CO and the formation of NO. As a result of the diffusion flame structure, the developed CO oxidation and NO formation chemical time scales are dependent only upon the inlet conditions to the burner.

Use of the developed characteristic time scales in designing a lower emissions disc combustor was also demonstrated. In particular for negligible fuel droplet effects, simple analysis of the correlating parameters showed an optimum geometry existed for minimum emissions. Thus the belief that any method of reducing CO emissions inherently increases NO_x emissions was refuted. As long as CO and NO_x emissions originate primarily in separate regions of the flame, the possibility exists of reducing both of these pollutants simultaneously through proper tailoring of the mixing processes. If, however, CO and NO_x emissions are related to a single region of the flame structure as was the case with the free stream flame, then the decrease in one of these pollutant species will indeed increase the other.

Correlations established for the axisymmetric disc-in-duct combustor have been extended to a conventional automotive gas turbine combustor for which heterogeneous effects are negligible. They are semi-empirical, requiring some emissions data for quantification, but have their origin in a more fundamental characteristic time model of the liquid fuel spray, turbulent diffusion flames found in conventional combustors (Mellor, 1976). Here we have attempted to show how such correlations can eliminate some of the cut-and-try development currently required for combustor design. It remains to develop similar techniques for more complicated, aircraft type combustors and to properly include fuel evaporation times should they prove significant. Armed with this capability,

it is hoped that the correlations will be used to define optimum emissions for each EPA engine class (EPA, 1973) and then compared with the existing standards to examine their stringency.

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Appendix A

Statistical Data Analysis

In the analysis of experimental data, there are many instances in which an increase in one variable is in general accompanied by an increase in another. Stated differently, large values of one variable tend to be associated with large values of a second and small values of the first tend to be associated with small values of the second, without the existence of a strict mathematical relationship between the two. In such a case, the variables are said to be correlated rather than functionally related. In this appendix, the least squares method of determining the "best" correlation is developed and the concept of a correlation coefficient to test the validity of the correlation is introduced. The development closely follows the analysis of Volk (1958).

If the pairs of values of the variables associated with each data point are designated x and y , with y assigned to the variable which is imprecisely known, a straight line through the data is expressed as

$$\hat{y} = a + bx \quad \text{A-1}$$

where $\hat{y}$ = estimated value of y for an observed x

a = intercept

b = slope of the line.

The values of a and b are determined such that the sum of the squares of the vertical deviations of the point from this line will be a minimum.

If there are N pairs of values (x_i, y_i) , the function to minimize is

$$\sum_{i=1}^N (y_i - \hat{y}_i)^2 = \sum_{i=1}^N (y_i - a - bx_i)^2 \quad \text{A-2}$$

Employing the usual methods of calculus, the above function can be partially differentiated with respect to a and then b and the subsequent

derivatives set equal to zero. Two simultaneous equations then result for the solution of a and b.

$$aN + b \sum_{i=1}^N x_i = \sum_{i=1}^N y_i \quad A-3$$

$$a \sum_{i=1}^N x_i + b \sum_{i=1}^N x_i^2 = \sum_{i=1}^N x_i y_i \quad A-4$$

Solving Equations A-3 and A-4 gives the desired values of a and b.

$$b = \frac{\frac{\sum_{i=1}^N x_i y_i}{N} - \frac{\sum_{i=1}^N y_i}{N} \frac{\sum_{i=1}^N x_i}{N}}{\frac{\sum_{i=1}^N x_i^2}{N} - \frac{\sum_{i=1}^N x_i}{N} \frac{\sum_{i=1}^N x_i}{N}} \quad A-5$$

$$a = \frac{\frac{\sum_{i=1}^N y_i}{N} - b \frac{\sum_{i=1}^N x_i}{N}}{1} = \bar{y} - b\bar{x} \quad A-6$$

Using these values of a and b, Equation A-1 predicts the most probable value of y, $\hat{y}_i$, for each x_i .

Although Equations A-5 and A-6 define the least squares straight line through any group of data points, it remains to develop a criterion to test whether or not it is reasonable to draw a straight line through the points. In other words, a straight line could be defined through a "shotgun" scatter plot, but the validity of the line would be questionable. Thus a criterion is needed to judge the validity of any linear correlation.

If no correlation between x_i and y_i exists, then the variation in the y values results only from error in the experimental reproducibility of the measurement. The magnitude of this error is given by the variance of the data.

$$\sigma^2 = \frac{1}{N} \sum_{i=1}^N (y_i - \bar{y})^2 \quad A-7$$

Since here the mean value of the y_i 's is the most probable one, the variance is seen to be just the average of the sum of the squares of deviation from the most probable y value.

If a correlation between x_i and y_i exists, then the randomness of the y values is decreased. Here a most probable y value $\hat{y}$, is predicted for each x_i value. The experimentally measured values of y may, however, be distributed about this most probable y value. Thus again a variance of the data can be defined.

$$\sigma^2 = \frac{1}{N} \sum_{i=1}^N (y_i - \hat{y}_i)^2 \quad \text{A-8}$$

The difference between Equations A-7 and A-8 represents the portion of the variance of the original data that has been removed by the linear correlation. After simplification, this difference can be expressed as

$$\frac{\frac{1}{N} \sum_{i=1}^N (y_i - \bar{y})^2}{\frac{1}{N} \sum_{i=1}^N (y_i - \hat{y}_i)^2} = \frac{b^2}{N} \left(\frac{\sum_{i=1}^N x_i^2}{1} - \frac{\sum_{i=1}^N x_i}{1} \frac{\sum_{i=1}^N x_i}{N} \right) \quad \text{A-9}$$

The first term in Equation A-9 is the error in the data if no correlation exists, and the second term is the error in the data if a linear correlation exists. The right hand side of Equation A-9 is then the error removed from the data if a correlation does exist. The ratio of the error removed by the correlation to the error in the original data is a measure of the goodness of the correlation. The square root of this ratio is denoted as the correlation coefficient.

$$r = \left[\frac{b^2 \left(\frac{\sum_{i=1}^N x_i^2}{1} - \frac{\sum_{i=1}^N x_i}{1} \frac{\sum_{i=1}^N x_i}{N} \right)}{\frac{1}{N} \sum_{i=1}^N (y_i - \bar{y})^2} \right]^{1/2} \quad \text{A-10}$$

When there is perfect correlation between x_i and y_i , there is no residual of y_i from $\hat{y}_i$ and the correlation coefficient is equal to unity. Conversely when there is no correlation and none of the sums of squares of deviation is removed by the linear relationship, the correlation coefficient equals zero. There is some probability that the correlation coefficient will be non-zero even if there is no correlation. However this probability diminishes as the number of data points increases due to the increased chance of a more random distribution. For example considering 10 data points, there is a 0.001 probability of getting an r value of 0.812 or larger when there is no correlation. If the number of data points is increased to 50, then the value of r decreases to 0.443 for a 0.001 probability. For the number of data points in excess of 50, a value of the correlation coefficient in excess of 0.8 can be considered a good correlation, and a value of the correlation coefficient in excess of 0.9 can be considered an excellent correlation. The calculated value of the correlation coefficient then serves as a criterion for deciding the goodness of the "best" fit straight line through the data.

After the validity of a correlation is verified, the variance of the values of y about the most probable y should also be determined since this gives an estimate of the remaining error in the experimental data. However in place of the variance, the standard deviation is commonly employed.

$$\sigma = \left[\frac{\sum (y_i - \hat{y}_i)^2}{N-2} \right]^{1/2} \quad \text{A-11}$$

The denominator of Equation A-11 is equal to the number of data points less the number of restrictions placed on the data. Since the least squares line is arrived at by minimizing the squared deviation of the data with respect to both a and b , two restrictions are placed on the data. On comparison of Equations A-8 and A-11, it can be noted that as the number of data points increases, the variance is just equal to the square of the standard deviation. Hence the variance and the standard deviation convey related information concerning the magnitude of the

scatter in the experimental measurements. It should also be noted that for a normal distribution 68% of the data fall within a standard deviation.

In summary, a straight line can be fit through any group of data points (x,y) via application of Equations A-5 and A-6 which determine the slope and intercept of the line, respectively. However the correlation coefficient of the determined line should also be calculated (Equation A-10) and used as a criterion to judge the goodness of the linear relationship. Finally, the standard deviation of the experimental data about the straight line should also be calculated (Equation A-11) since it is a measure of the uncertainty remaining in the data after correlation.

Appendix B

Alternate Method of Correlating
Jet-A Emissions Data

In the discussion of Jet-A emissions data of Section IV, a simple but physically motivated functional dependence of τ_{eb} was incorporated into the previously developed parameters for use in correlating emissions from burners in which heterogeneous effects are negligible. At that time, it was noted that the primary effect of large fuel droplets upon the flame structure was the penetration of droplets through the shear layer and into the free stream. In particular as the fuel droplet lifetime increased (greater droplet penetration distance), the NO_x emissions were found to decrease while CO emissions increased. In the following, the magnitude of τ_{eb} will be used in a more explicit manner to determine the extent of the fuel droplet penetration into the flame structure and thus the resulting influence upon burner emissions.

Recall from Fig. 3-4 that depending upon the extent of fuel droplet penetration into the free stream, the flame structure may or may not exhibit a free stream flame zone. For the cases where a free stream flame zone does occur, the total NO_x burner emissions can be considered to be a sum of the shear layer flame emissions and the free stream flame emissions.

$$\text{NO}_x \text{EI} \sim a \frac{\tau_{sl, \text{no}}}{\tau_{\text{no}}} + b \frac{\tau_{fs, \text{no}}}{\tau_{\text{no}}} \quad \text{B-1}$$

where a is the fraction of the fuel flow vaporized within the shear layer and b is the fraction of the fuel flow penetrating into the free stream. The first term on the right of Equation B-1 has been shown to correlate NO_x emissions from the shear layer flame and the second term is an analogous expression for correlation of NO_x emissions from the free stream flame. The importance of each of these two flames to the total

burner NO_x emissions is determined by the fraction of the fuel flow consumed in each flame. Note that for the case of no droplet penetration into the free stream, b is zero ($a = 1$) and Equation B-1 reduces to the expression used to correlate the NO_x emissions from the propane fueled flames.

Equation B-1 can be rewritten in the more convenient form

$$\text{NO}_x \text{EI} \sim \frac{\tau_{sl,no}}{\tau_{no}} \left(a + b \frac{\tau_{fs,no}}{\tau_{sl,no}} \right) \quad \text{B-2}$$

Since the Jet-A data are not too different from the propane data, the free stream flame does not dominate the overall flame structure but only represents a perturbation upon the case where droplets are negligible. Thus most of the fuel continues to be consumed by the shear layer flame and the fuel fraction a is considerably larger than the fraction b . In addition, the convective velocity in the region of the free stream flame is considerably greater than that of the shear layer near the combustor centerline. If the turbulent velocity, u' , is but a similar percentage of the convective velocity through each of these two regions of the burner, then the mixing (quenching) time for NO formation in the free stream is considerably shorter than $\tau_{sl,no}$. Since $a > b$ and $\tau_{fs,no} < \tau_{sl,no}$, the second term within the parenthesis of Equation B-2 can be considered as a first approximation to be negligible in comparison to a . Hence

$$\text{NO}_x \text{EI} \sim a \frac{\tau_{sl,no}}{\tau_{no}} \quad \text{B-3}$$

which suggests that significant NO formation occurs only in the shear layer flame structure. The total burner NO_x emissions then decrease as droplet penetration into the free stream increases (i.e., a decreases). However before Equation B-3 can be used to correlate Jet-A emissions, a means of estimating the fuel fraction a is needed.

Consider the time of flight, t_f required for a fuel droplet to travel from the fuel nozzle to the free stream. Neglecting drag effects,

this time can be estimated as the distance from the nozzle to the free stream divided by the initial droplet velocity, V_f . The distance of interest will depend upon the size of the recirculation zone behind the disc (see Fig. 3-4). Following Winterfeld (1965), the length of the recirculation zone scales linearly with the disc diameter; consequently, here the distance from the nozzle to the free stream is taken as the disc diameter. An empirical expression for the initial velocity of droplets produced by a simplex pressure atomizer is given by Crowe (1974) such that the time of flight for a droplet to travel from the nozzle to the free stream becomes

$$t_f = \frac{d}{V_f} = \frac{d}{14.4 \sqrt{\Delta P}} \quad B-4$$

where ΔP is the pressure drop across the nozzle (atm) and V_f has units of m/sec. The time of flight in comparison to τ_{eb} can be used as a criterion for determining the percentage of fuel which evaporates within the recirculation zone.

The physical relationship governing droplet evaporation was given previously as Equation 2-1 and is rewritten below in terms of the mass of the droplet, M , at time t normalized by the initial mass of the fuel droplet, M_0 .

$$\frac{M}{M_0} = \left(1 - \frac{\beta t}{d_0^2} \right)^{3/2} = \left(1 - \frac{t}{\tau_{eb}} \right)^{3/2} \quad B-5$$

where the definition of τ_{eb} (Equation 2-2) has been utilized. Equation B-5 then gives the fraction of the initial mass of a fuel droplet remaining to be evaporated as a function of time. Note that in Equation B-5, the time has an upper bound of τ_{eb} at which point the droplet is completely evaporated.

The fraction of the fuel penetrating into the free stream, b , can now be taken as the fraction of the initial mass of the droplet remaining after a time t_f .

$$b = \frac{M_{t=t_f}}{M_O} = \left[1 - \frac{t_f}{\tau_{eb}} \right]^{3/2} \quad \text{B-6}$$

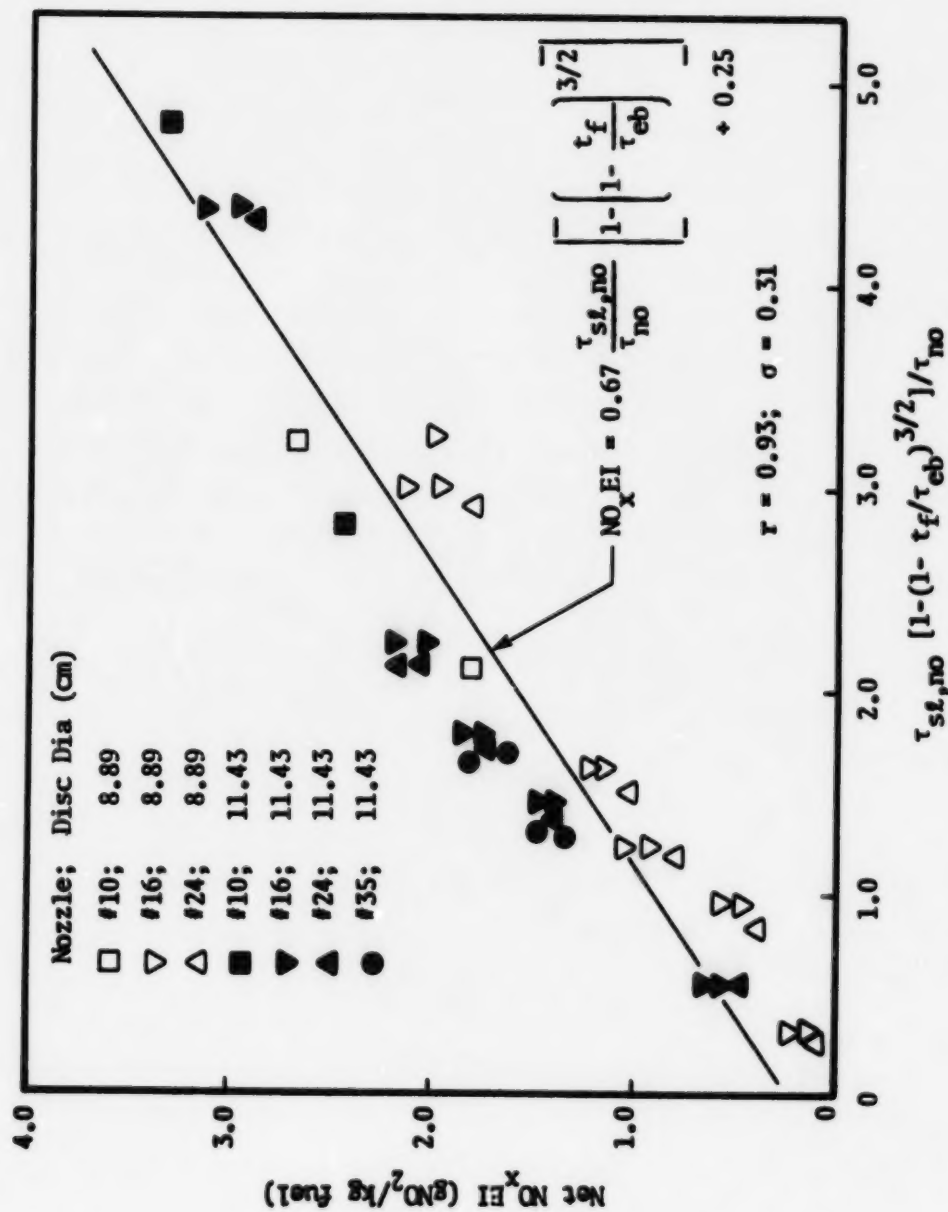
Since a and b must sum to unity, the correlating expression for the disc burner NO_x emissions becomes

$$\text{NO}_x \text{EI} \sim \frac{\tau_{sl, \text{no}}}{\tau_{\text{no}}} \left[1 - \left[1 - \frac{t_f}{\tau_{eb}} \right]^{3/2} \right] \quad \text{B-7}$$

Note that if $t_f \geq \tau_{eb}$, then the quantity within the brackets of Equation B-7 reduces to unity and no effect of τ_{eb} remains in the correlating parameter.

As a demonstration of the success of this alternate expression for correlating Jet-A emissions, the data contained in Fig. 4-9 are replotted in Fig. B-1 against Equation B-7. Note that with the inclusion of τ_{eb} , the correlation coefficient has increased from 0.91 in Fig. 4-9 to 0.93 in Fig. B-1. In addition, the slope of the line in Fig. B-1 is nearly identical to that of the line for the propane data shown in Fig. 4-2. Thus the use of Equation B-7 to correlate NO_x emissions effectively reduces the Jet-A fuel emissions to those when propane fuel was burned.

Although Equation B-7 is equally as successful as Equation 4-12 in correlating Jet-A emissions measurements from the present investigation, the former equation is more peculiar to the present experimental burner. Thus in correlating pollutant emissions from other diffusion flame combustion devices, the use of Equation 4-12 is preferred.

Figure B-1. Alternate correlation of Jet-A fuel NO_x emissions

Appendix C

Data Listing

All of the experimental data obtained and analyzed in the present investigation are included in the tables of this Appendix. The first five tables contain the data for the three circular disc flameholders when the burner was fueled with propane. Tables C-6 through C-9 also contain data from propane fueled flames but for toothed disc flameholders. Data corresponding to the various fuel nozzles and disc diameters utilized in the Jet-A experiments are contained in Tables C-10 through C-16. In addition to the emissions listed in each table, the fuel nozzle differential injection pressure is also given.

Table C-1

Emissions Data: P, $\dot{m}$, Fuel, 6.09 cm Diameter Disc, #16 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T _i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO _x $\left[\frac{\text{gNO}_x}{\text{kgfuel}} \right]$	CO+H $\left[\frac{\text{gCO}}{\text{kgfuel}} \right]$	HCl+H $\left[\frac{\text{gHCl}}{\text{kgfuel}} \right]$
1.0	4.0	500	0.2	21.0	1.06	66.2	18.4
1.0	4.0	500	0.2	21.0	0.91	68.7	18.4
1.0	5.0	500	0.2	19.5	1.27	59.5	15.5
1.0	5.0	500	0.2	19.5	1.28	55.1	15.6
1.0	8.0	500	0.2	17.2	2.01	35.4	14.4
1.0	8.0	500	0.2	17.2	1.95	36.2	12.0
0.75	6.8	500	0.2	13.0	2.24	34.6	8.5
0.75	6.8	500	0.2	13.0	2.29	35.4	9.6

Table C-2

Emissions Data: Propane Fuel, 8.89 cm Diameter Disc, #35 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T_i (°K)	t_o	ΔP (Nozzle) (atm)	Net NO _x EI $\left[\frac{\text{gNO}_2}{\text{kg fuel}} \right]$	COEI $\left[\frac{\text{gCO}}{\text{kg fuel}} \right]$	HCEI $\left[\frac{\text{gCl}_4}{\text{kg fuel}} \right]$
1.0	5.0	500	0.2	9.4	1.89	57.7	7.8
1.0	5.0	500	0.2	9.4	1.64	55.3	13.0
1.5	5.0	500	0.2	13.2	0.93	38.5	11.5
1.5	5.0	500	0.2	13.2	1.03	45.5	13.0
1.0	2.0	500	0.2	12.8	0.62	90.0	43.4
1.0	2.0	500	0.2	12.8	0.77	107.9	59.0
1.0	8.0	500	0.2	6.6	2.14	32.2	14.7
1.0	8.0	500	0.2	6.6	2.20	35.7	14.6
1.0	5.0	500	0.1	6.7	2.61	57.3	26.9
1.0	5.0	500	0.1	6.7	2.14	67.5	52.9
1.0	5.0	500	0.3	13.1	0.80	66.9	28.9
1.0	5.0	500	0.3	13.1	0.92	69.9	24.6
1.0	5.0	450	0.2	9.4	1.52	44.5	12.1
1.0	5.0	450	0.2	9.4	1.64	58.1	15.6
1.0	5.0	550	0.2	9.4	1.77	45.5	7.8
1.0	5.0	550	0.2	9.4	2.08	-----	-----

Table C-3

Emissions Data: Propane Fuel, 11.43 cm Diameter Disc, #16 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T_i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO_x EI $\left[\frac{gNO_2}{kgfuel} \right]$	COEI $\left[\frac{gCO}{kgfuel} \right]$	HCEI $\left[\frac{gCH_4}{kgfuel} \right]$
0.5	5.0	500	0.2	14.8	3.36	29.3	2.6
1.0	5.0	500	0.2	19.5	2.44	64.6	23.8
1.5	5.0	500	0.2	33.4	1.36	108.7	58.5
1.0	2.0	500	0.2	22.9	0.86	154.3	99.7
1.0	8.0	500	0.2	17.2	3.11	29.3	8.7
1.0	5.0	450	0.2	20.1	2.22	69.8	29.5
1.0	5.0	550	0.2	18.6	2.34	63.7	21.7

Table C-4

Emissions Data: Propane Fuel, 11.43 cm Diameter Disc, #24 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T _i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO _x $\left(\frac{\text{gNO}_2}{\text{kg fuel}} \right)$	COEI $\left(\frac{\text{gCO}}{\text{kg fuel}} \right)$	HCEI $\left(\frac{\text{gCH}_4}{\text{kg fuel}} \right)$
0.5	5.0	500	0.2	9.8	3.80	36.9	4.3
1.0	5.0	500	0.2	11.7	2.86	49.4	17.3
1.5	5.0	500	0.2	17.1	2.05	87.4	21.7
1.0	2.0	500	0.2	14.6	0.62	159.6	91.0
1.0	8.0	500	0.2	9.7	3.92	19.8	4.3
1.0	5.0	450	0.2	11.0	3.03	54.7	15.2
1.0	5.0	550	0.2	12.4	2.43	49.4	16.5

Table C-5

Emissions Data: Propane Fuel, 11.43 cm Diameter Disc, #35 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T_i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO_x $\left[\frac{gNO_2}{kgfuel} \right]$	COEI $\left[\frac{gCO}{kgfuel} \right]$	HClEI $\left[\frac{gHCl}{kgfuel} \right]$
1.0	5.0	500	0.2	9.4	1.98	76.0	31.2
1.0	5.0	500	0.2	9.4	2.39	75.4	29.5
1.5	5.0	500	0.2	13.2	1.20	98.8	49.9
1.5	5.0	500	0.2	13.2	1.13	101.8	60.7
1.0	2.0	500	0.2	12.8	0.71	177.8	99.7
1.0	2.0	500	0.2	12.8	0.81	154.3	96.0
1.0	8.0	500	0.2	6.6	3.50	58.1	23.4
1.0	8.0	500	0.2	6.6	2.80	57.2	28.2
1.0	5.0	500	0.1	6.7	3.06	66.0	19.5
1.0	5.0	500	0.1	6.7	2.86	61.4	19.1
1.0	5.0	500	0.3	13.1	1.83	67.9	27.5
1.0	5.0	500	0.3	13.1	1.60	67.9	21.7
1.0	5.0	450	0.2	9.4	1.77	73.0	32.1
1.0	5.0	450	0.2	9.4	2.19	72.6	31.2
1.0	5.0	550	0.2	9.4	1.88	72.6	26.9
1.0	5.0	550	0.2	9.4	2.03	67.1	26.0

Table C-6

Emissions Data: Propane Fuel, 8 Teeth Disc, #16 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T_i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO _x EI $\left[\frac{\text{gNO}_2}{\text{kgfuel}} \right]$	COEI $\left[\frac{\text{gCO}}{\text{kgfuel}} \right]$	HCEI $\left[\frac{\text{gCH}_4}{\text{kgfuel}} \right]$
1.0	2.0	500	0.2	22.9	0.47	102.7	52.8
1.0	2.0	500	0.2	22.9	0.58	99.5	45.8
1.0	5.0	500	0.2	19.5	1.71	29.5	4.8
1.0	5.0	500	0.2	19.5	1.15	32.4	5.7
1.0	8.0	500	0.2	17.2	2.56	20.2	9.2
1.0	8.0	500	0.2	17.2	2.50	26.0	8.8
0.75	6.8	500	0.2	15.0	----	20.8	10.9
0.75	6.8	500	0.2	15.0	2.59	22.7	14.4
1.5	5.0	500	0.2	35.4	----	46.5	12.0
1.5	5.0	500	0.2	35.4	1.55	51.0	18.4

Table C-7

Emissions Data: Propane Fuel, 23 Teeth Disc, #16 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T_i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO _x EI $\left(\frac{\text{gNO}_2}{\text{kgfuel}} \right)$	COEI $\left(\frac{\text{gCO}}{\text{kgfuel}} \right)$	HCEI $\left(\frac{\text{gCH}_4}{\text{kgfuel}} \right)$
1.0	2.0	500	0.2	22.9	0.58	120.2	40.4
1.0	2.0	500	0.2	22.9	0.57	132.4	43.7
1.0	5.0	500	0.2	19.5	1.53	28.8	10.5
1.0	5.0	500	0.2	19.5	1.64	29.5	17.9
1.0	8.0	500	0.2	17.2	2.68	21.5	9.8
1.0	8.0	500	0.2	17.2	2.85	26.0	11.4
0.75	6.8	500	0.2	13.0	2.62	20.2	14.0
0.75	6.8	500	0.2	13.0	2.56	20.8	14.4
1.5	5.0	500	0.2	33.4	1.35	55.1	21.4
1.5	5.0	500	0.2	33.4	1.27	57.2	22.3

Table C-8

Emissions Data: Propane Fuel, 13 Teeth Disc, #16 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T_i (°K)	ϕ_o	ΔP (Nozzle) (atm)	$N_c \sim NO_x EI$ $\left(\frac{gNO_2}{kgfuel} \right)$	COEI $\left(\frac{gCO}{kgfuel} \right)$	HCEI $\left(\frac{gCH_4}{kgfuel} \right)$
1.0	2.0	500	0.2	22.9	0.81	184.3	91.9
1.0	2.0	500	0.2	22.9	0.62	172.3	85.3
1.0	5.0	500	0.2	19.5	2.03	41.9	9.8
1.0	5.0	500	0.2	19.5	2.10	38.6	8.1
1.0	8.0	500	0.2	17.2	3.40	20.2	4.8
1.0	8.0	500	0.2	17.2	3.42	17.1	3.1
0.75	6.8	500	0.2	13.0	3.71	18.3	2.4
0.75	6.8	500	0.2	13.0	3.70	20.2	2.3
1.5	5.0	500	0.2	33.4	1.45	85.4	45.9
1.5	5.0	500	0.2	33.4	1.53	80.3	35.9

Table C-9
Emissions Data: Propane Fuel, 29 Teeth Disc, #16 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T_i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO_x $\left[\frac{gNO_2}{kg fuel} \right]$	COEI $\left[\frac{gCO}{kg fuel} \right]$	HCEI $\left[\frac{gH_4}{kg fuel} \right]$
1.0	2.0	500	0.2	22.9	0.56	234.0	109.3
1.0	2.0	500	0.2	22.9	0.86	234.1	105.0
1.0	5.0	500	0.2	19.5	2.20	88.0	45.9
1.0	5.0	500	0.2	19.5	2.23	73.3	40.3
1.0	8.0	500	0.2	17.2	3.54	35.4	14.2
1.0	8.0	500	0.2	17.2	3.54	23.4	8.8
0.75	6.8	500	0.2	13.0	3.76	27.4	5.5
0.75	6.8	500	0.2	13.0	3.68	29.5	7.0
1.5	5.0	500	0.2	33.4	----	----	----
1.5	5.0	500	0.2	33.4	1.59	136.7	98.4

Table C-10

Emissions Data: Jet-A Fuel, 8.89 cm Diameter Disc, #10 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T _i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO _x EI $\left(\frac{\text{gNO}_2}{\text{kg fuel}} \right)$	COEI $\left(\frac{\text{gCO}}{\text{kg fuel}} \right)$	HCEI $\left(\frac{\text{gCH}_4}{\text{kg fuel}} \right)$
1.0	5.0	500	0.1	8.9	1.84	63.3	36.0
0.75	6.8	500	0.2	30.6	2.68	47.1	26.3

Table C-11

Emissions Data: Jet-A Fuel, 8.89 cm Diameter Disc, #16 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T _i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO _x EI $\left(\frac{\text{gNO}_2}{\text{kgfuel}} \right)$	COEI $\left(\frac{\text{gCO}}{\text{kgfuel}} \right)$	HCEI $\left(\frac{\text{gCH}_4}{\text{kgfuel}} \right)$
1.0	2.0	500	0.2	7.5	0.13	167.0	92.6
1.0	2.0	500	0.2	7.5	0.16	157.0	86.4
1.0	5.0	500	0.2	7.8	1.22	93.6	63.8
1.0	5.0	500	0.2	7.8	1.17	90.8	57.6
1.0	8.0	500	0.2	7.7	2.13	68.2	39.1
1.0	8.0	500	0.2	7.7	1.95	66.8	39.5
1.0	5.0	500	0.3	17.2	1.05	76.1	40.2
1.0	5.0	500	0.3	17.2	0.93	76.1	40.2
1.5	5.0	500	0.2	17.2	0.49	96.8	55.7
1.5	5.0	500	0.2	17.2	0.47	99.9	57.8
0.75	6.8	500	0.2	4.0	1.95	69.4	41.2

Table C-12

Emissions Data: Jet-A Fuel, 8.89 cm Diameter Disc, #24 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T_i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO _x EI $\left[\frac{\text{gNO}_2}{\text{kgfuel}} \right]$	COEI $\left[\frac{\text{gCO}}{\text{kgfuel}} \right]$	HCEI $\left[\frac{\text{gCH}_4}{\text{kgfuel}} \right]$
1.0	2.0	500	0.2	5.5	0.11	146.7	70.0
1.0	5.0	500	0.2	5.5	1.01	85.4	57.6
1.0	8.0	500	0.2	5.6	1.75	62.9	47.3
1.0	5.0	500	0.3	11.0	0.78	78.5	54.0
1.5	5.0	500	0.2	11.0	0.37	88.2	57.8

Table C-13

Emissions Data: Jet-A Fuel, 11.43 cm Diameter Disc, #10 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T_i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO _x EI $\left[\frac{\text{gNO}_2}{\text{kgfuel}} \right]$	COEI $\left[\frac{\text{gCO}}{\text{kgfuel}} \right]$	HCEI $\left[\frac{\text{gCH}_4}{\text{kgfuel}} \right]$
1.0	5.0	500	0.2	8.9	2.48	40.1	14.3
0.75	6.8	500	0.2	30.6	3.52	35.5	10.3

Table C-14

Emissions Data: Jet-A Fuel, 11.45 cm Diameter Disc, #16 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T_i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO_x EI $\left[\frac{gNO_2}{kgfuel} \right]$	COEI $\left[\frac{gCO}{kgfuel} \right]$	HCEI $\left[\frac{gHI_1}{kgfuel} \right]$
1.0	2.0	500	0.2	7.5	0.60	228.0	136.0
1.0	2.0	500	0.2	7.5	0.61	244.0	119.0
1.0	5.0	500	0.2	7.8	2.05	60.1	29.6
1.0	5.0	500	0.2	7.8	2.18	60.1	26.3
1.0	8.0	500	0.2	7.7	2.97	45.3	22.6
1.0	8.0	500	0.2	7.7	3.11	37.1	17.1
1.0	5.0	500	0.3	17.2	1.77	61.1	26.3
1.0	5.0	500	0.3	17.2	1.86	61.1	26.1
1.5	5.0	500	0.2	17.2	1.42	80.5	47.5
1.5	5.0	500	0.2	17.2	1.46	73.4	43.3

Table C-15
Emissions Data: Jet-A Fuel, 11.43 cm Diameter Disc, #24 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T_i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO_x $\left[\frac{gNO_2}{kgfuel} \right]$	COEI $\left[\frac{gCO}{kgfuel} \right]$	HCEI $\left[\frac{gCl_4}{kgfuel} \right]$
1.0	2.0	500	0.2	5.5	0.50	192.0	105.0
1.0	5.0	500	0.2	5.5	2.12	62.5	27.2
1.0	5.0	500	0.2	5.5	2.18	60.1	50.5
1.0	8.0	500	0.2	5.6	2.92	57.8	16.5
1.0	5.0	500	0.5	11.0	1.84	69.2	27.7
1.5	5.0	500	0.2	11.0	1.46	80.5	45.5

Table C-16

Emissions Data: Jet-A Fuel, 11.43 cm Diameter Disc, #35 Nozzle

$\dot{m}$ (kg/sec)	P (atm)	T_i (°K)	ϕ_o	ΔP (Nozzle) (atm)	Net NO_xEI $\left(\frac{gNO_2}{kgfuel} \right)$	COEI $\left(\frac{gCO}{kgfuel} \right)$	HCEI $\left(\frac{gCH_4}{kgfuel} \right)$
1.0	5.0	500	0.3	3.7	1.64	73.7	30.5
1.0	5.0	500	0.3	3.7	1.82	65.1	27.7
1.5	5.0	500	0.2	3.7	1.43	85.6	45.4
1.5	5.0	500	0.2	3.7	1.47	83.0	39.6